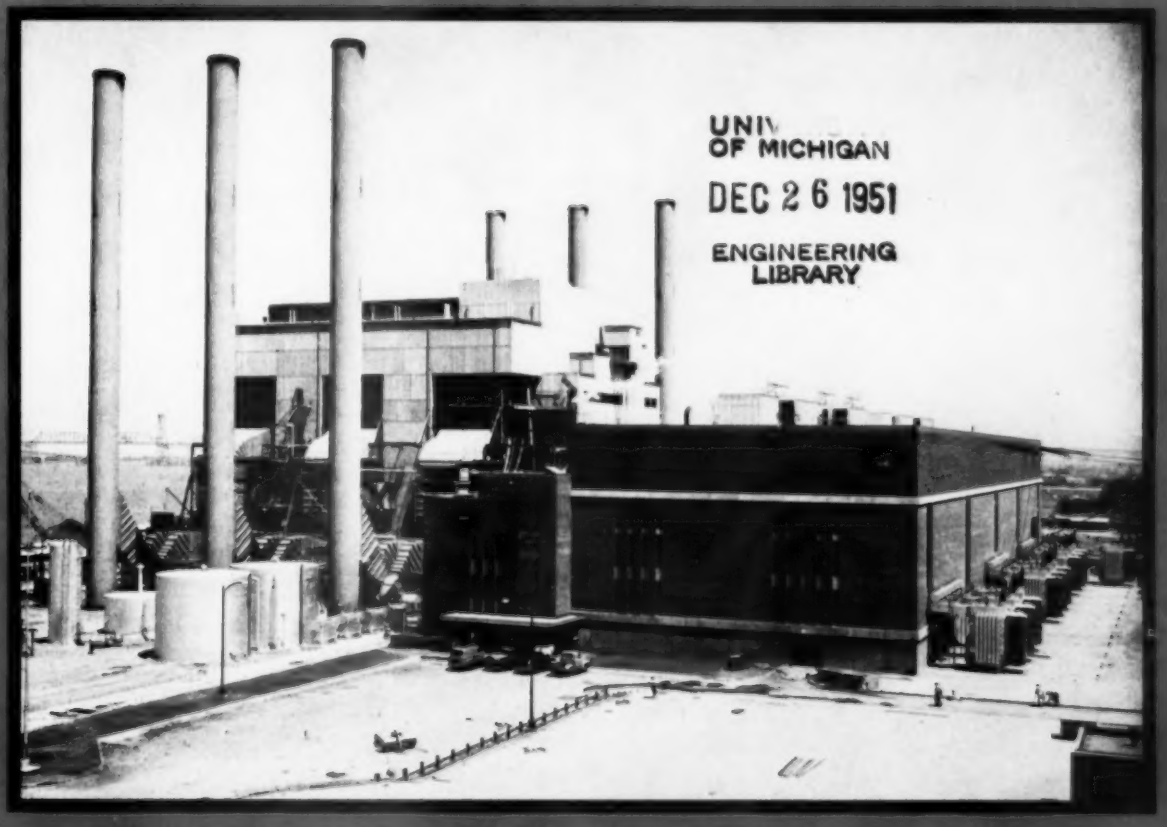
COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

RR

December 1951



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Contra Costa Steam Plant of Pacific Gas & Electric Company; see p. 34

Contra Costa Steam Plant ▸

Trends in Deaerating Heaters ▸

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C-E REHEAT BOILERS

CONTRA COSTA STEAM PLANT

PACIFIC GAS & ELECTRIC COMPANY

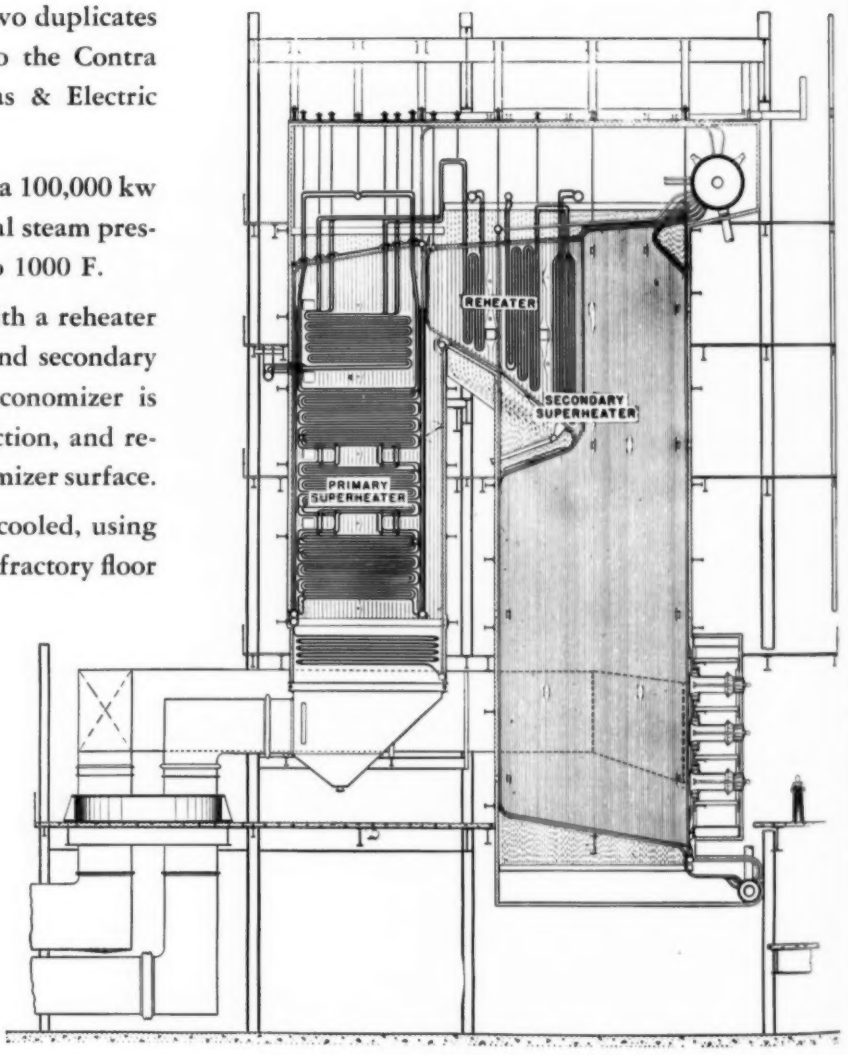
THE C-E Unit shown here is one of two duplicates now on order for the extension to the Contra Costa Steam Plant of the Pacific Gas & Electric Company.

Each of the units is designed to serve a 100,000 kw turbine generator operating at an initial steam pressure of 1450 psi at 1000 F, reheated to 1000 F.

The units are of the radiant type with a reheater section located between the primary and secondary superheater surface. A finned tube economizer is located below the rear superheater section, and regenerative air heaters follow the economizer surface.

The furnaces are completely water cooled, using closely spaced plain tubes and have a refractory floor supported by water tubes.

Initially these units will be fired by oil or gas, but the design contemplates the possible future use of pulverized coal if desired. B-515



C-E

COMBUSTION ENGINEERING–SUPERHEATER, INC.

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ALL TYPES OF BOILERS, FURNACES, PULVERIZED FUEL SYSTEMS AND STOKERS; ALSO SUPERHEATERS, ECONOMIZERS AND AIR HEATERS

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

Vol. 23 **No. 6**

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ASME Annual Meeting at Atlantic City

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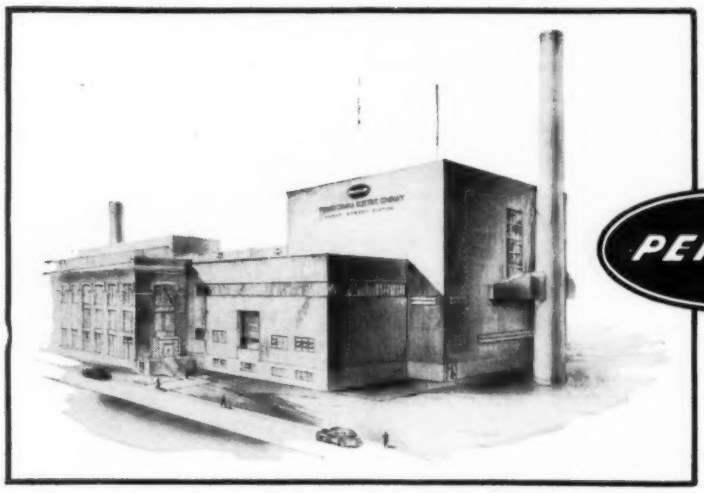
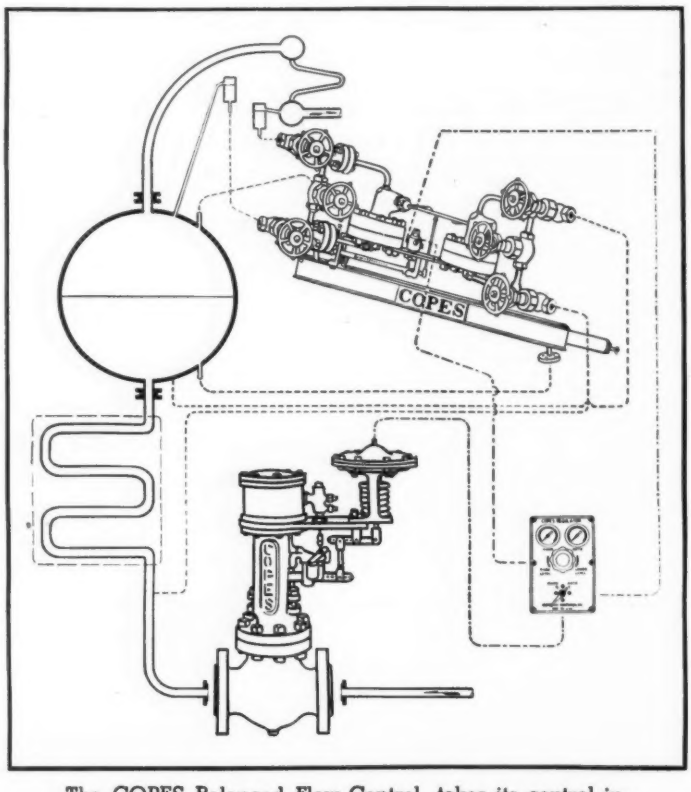
PENNSYLVANIA ELECTRIC CO. Places 6th order for COPES Multi-Element REGULATORS



Front Street addition includes two 1250-psi, 950-F C-E boilers, each rated at 400,000 pounds of steam per hour.



The COPES Balanced Flow Control, takes its control influences from steam flow, feed water flow and water level. Plant personnel can handle all maintenance easily; no need for "experts." The feed valve will stand up under *unlimited* pressure drops.

PENELEC

COPES Balanced Flow Control will be used for the 1952 Addition to Front Street Station of Pennsylvania Electric Company at Erie, Pennsylvania. This is the sixth time Penelec has ordered COPES Multi-Element Feed Water Control.

First installation was on the two original 675-psi boilers at Front Street, using two-element COPES Flowmatics. Two later orders placed Flowmatics on four 675-psi boilers at Seward. Two other orders equipped the four 875-psi boilers at Warren with COPES Flowmatics.

Now, for the Front Street Extension, Penelec has selected the COPES Balanced Flow—with control influenced by steam flow, feed flow and water level. This three-influence control is entirely independent of all other instruments and controls, and can remain on fully-automatic when they are out of service for routine inspection and care.

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COPES

BOILER FEED WATER REGULATION

COMBUSTION

Editorials

Reheat Comes of Age

Three years ago at the ASME Annual Meeting a series of papers on the reheat cycle for central stations was greeted with a mixture of intense interest and healthy skepticism. Too complicated, limited to base load plants, inflexible to rapidly changing operating conditions, increased thermal efficiency not commensurate with additional required investment—these were some of the laments of those who lacked confidence in or needed selling on the returns to be gained from adoption of the reheat cycle.

To the proponents of reheat—those who have made continuous use of it since the 1920's and those who became converted to it as one antidote to rising fuel costs—the papers on reheat operating experience presented at the 1951 ASME Annual Meeting just concluded at Atlantic City must have been a source of considerable satisfaction. Most of the fears expressed three years ago have proved groundless, and the utility industry both here and abroad has widely adopted reheat for installations starting at about 60,000 kw. One turbine manufacturer expects that reheat will account for 60.6 and 71.8 per cent of the total rating of large 3600-rpm machines to be shipped in 1952 and 1953, respectively.

This accomplished trend toward reheat did not come about haphazardly. The reports at Atlantic City show that the units have done their job well, but behind them are the designers who have proved that reheat need not be complicated, the consulting engineers who have recommended its adoption, and the managers of the utility industry who have backed up these recommendations with millions of dollars in new station investments.

That reheat has come of age marks another step in the progress of the power industry.

Is This the Next Step?

Coincident with the intensive centralization of controls in large steam-electric stations there has been a marked increase in the amount of operating and performance data collected. If this information is to serve a more useful purpose than mere accumulation of records, then it must be analyzed, correlated and tabulated. Inevitably this involves calculations, some routine and some quite complex.

This poses the question: will economies achieved by reduction in operating personnel through centralization be counterbalanced by an increase in the size of the results department to calculate and interpret the data?

In recent years much has been heard about mechanical brains equipped with memory devices. Also comments have been made that operators may be approaching the limits of their capacity to absorb the intelligence transmitted to the central control room, particularly in times of emergency. Viewed in this light it has been suggested that a logical future step in control room design would be the incorporation of some type of automatic computers capable of carrying out routine calculations and combining data into a limited number of categories of direct use to the operators.

When one considers the complex problems that have been entrusted to and solved by modern computing machines, it does not tax the imagination too much to expect that they will become a part of centralized controls in future power plants.

Instruments and Controls

Progress in many of the process industries would have been seriously impeded, and many notable achievements precluded, if development of controls and special instruments had not only kept pace with current requirements but often actually opened the way to further advances. The same may be said of the field of power generation.

It is inconceivable that such advances as are typified by the modern central station would have been attainable or even approached, without the means of maintaining close control of combustion, steam conditions, safety, reliable operation, plant performance and many other factors. Moreover, to mention one, with the trend toward centralized control, it is possible to minimize operating labor costs.

Differing principally in degree, and with certain limitations, this applies also to those smaller plants whose managements are alert to the opportunities offered. Of course, there are still too many small plants that either do not have the minimum of controls and instruments necessary to efficient operation, or do not make intelligent use of such as they have. This presents a difficult educational problem that has long existed.

However, it is encouraging to note the interest shown in boiler instrumentation at the recent meeting in Philadelphia, as reported elsewhere in this issue. It is through such meetings that this important branch of power engineering will achieve the prominence it deserves.

Contra Costa Steam Plant of the Pacific Gas & Electric Company

This new steam-electric generating station, with an initial capacity of 340,000 kw, is located in the San Francisco Bay region. It, at present, contains three main turbine-generators each served by two boilers at 1300 psig, 950 F. An extension, now under construction, will bring the total capacity up to 570,000 kw.

DESIGNED primarily to meet a remarkable growth in the industrial and residential loads of the San Francisco Bay region, but capable of supplying power over a large part of the Company's system through seven substations, the new Contra Costa steam-electric generating station of the Pacific Gas & Electric Company was formally dedicated late in August.

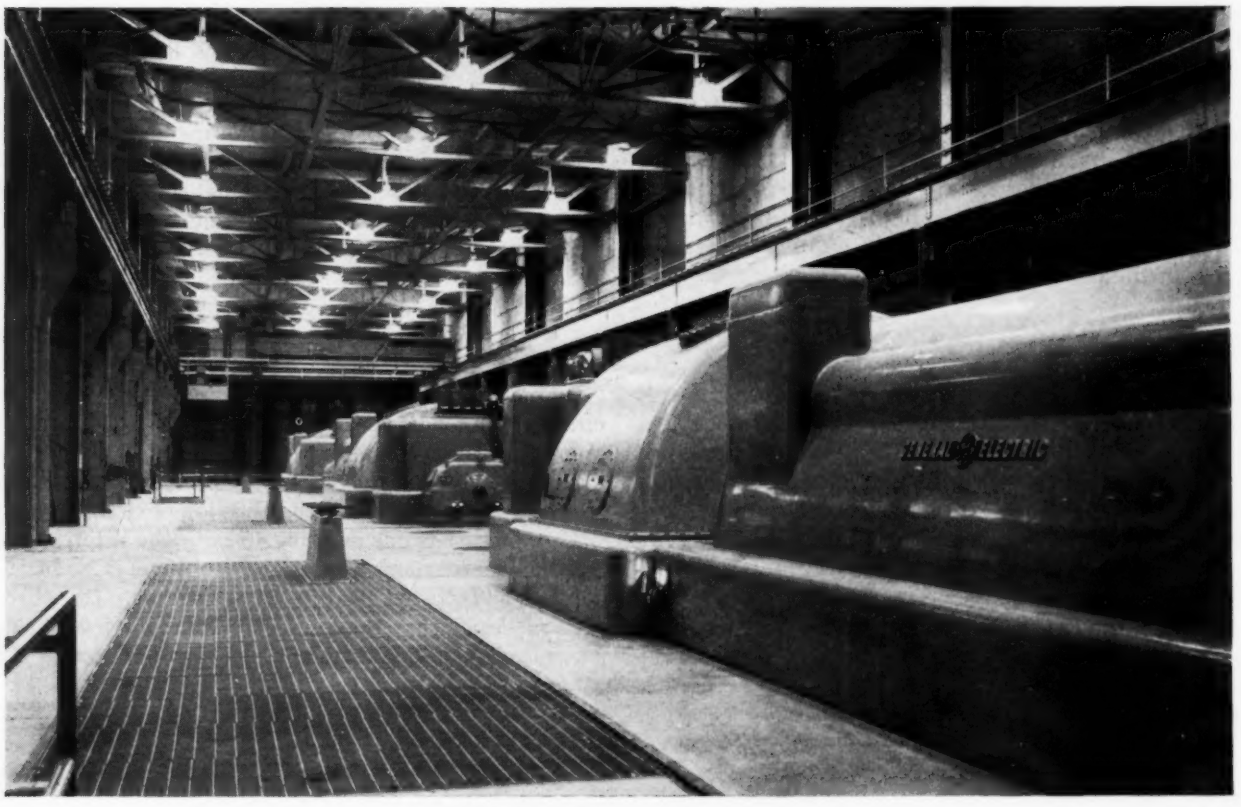
The gross normal operating capacity of the station, including three main units and three house units, is 340,000 kw. Each main unit and related house turbine are supplied with steam at 1300 psig, 950 F by two boilers. An extension is now in the construction stage, involving two additional units of like capacity whose operation is scheduled for the spring of 1953. However, these will employ the reheat cycle and each will be served by a single boiler, with the house units omitted. This will increase the total gross capacity to 570,000 kw.

A feature of the design is the loading flexibility, from very low loads on standby to almost full load in a matter of seconds. This provision is met, in part, by full-speed operation of the boiler feed pumps and draft fans, and by keeping all fuel burners in continuous operation. Centralized control has also been incorporated.

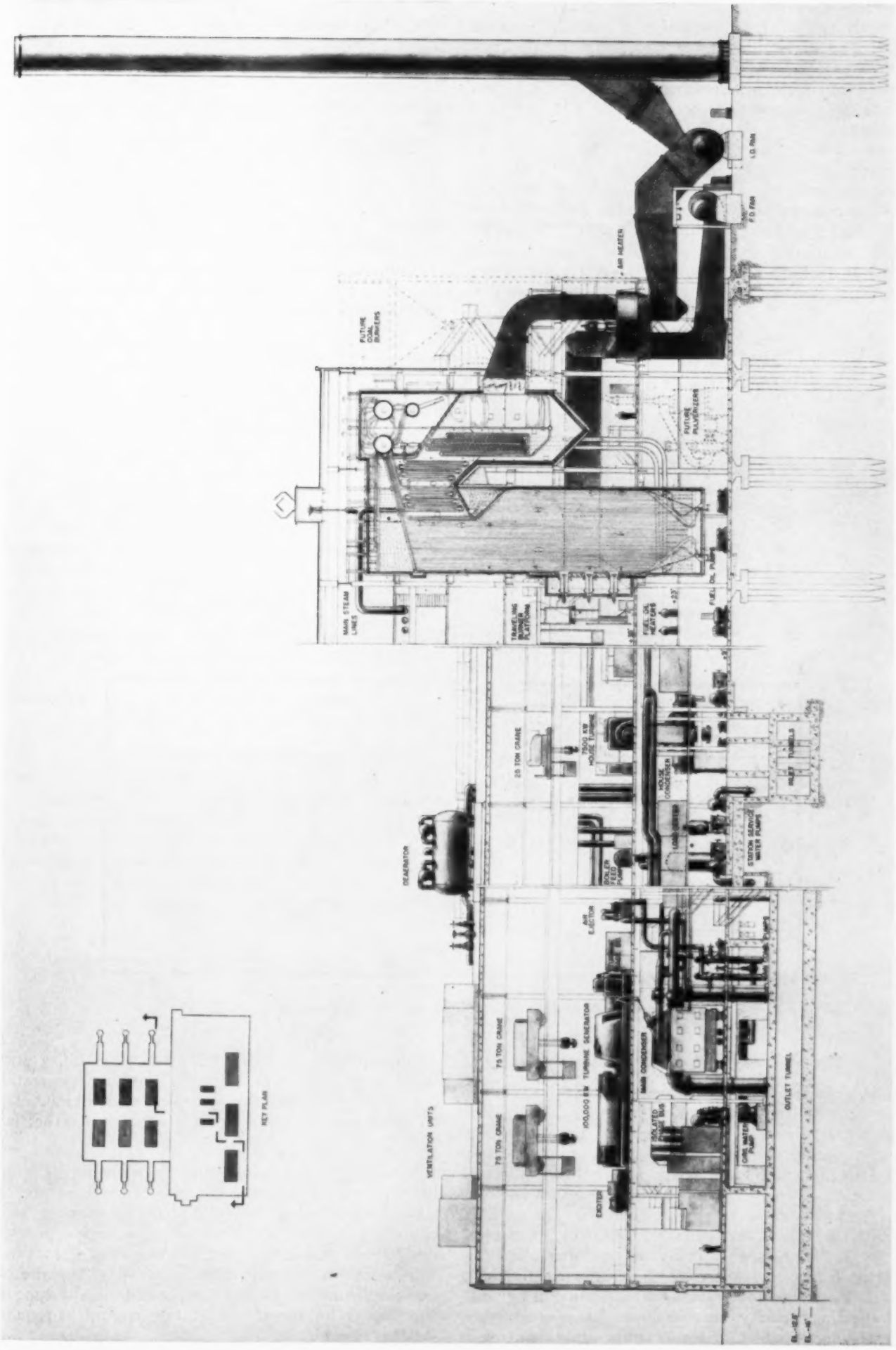
The site, near the mouth of the San Joaquin River and east of the city of Antioch, makes possible delivery of fuel by pipe line, rail, or water, in addition to providing ample condenser circulating water. Utility water for general plant use is taken from the Antioch mains.

Although the station was laid out to burn gas or oil initially, provision has been made for adapting it to burn pulverized coal with no reduction in steam output, if this should later become desirable.

The boiler house is of semi-outdoor design, but the turbine, auxiliary and control bays are completely housed in red brick with colored concrete trim. However, there is a steel-supported canopy over the boilers and certain working areas are protected by corrugated transite siding. The firing aisle is completely enclosed. All boiler areas and the stacks are carried on precast, reinforced concrete piling in order to minimize deflections, and the main



Main turbine-generators



Cross-sectional view through plant

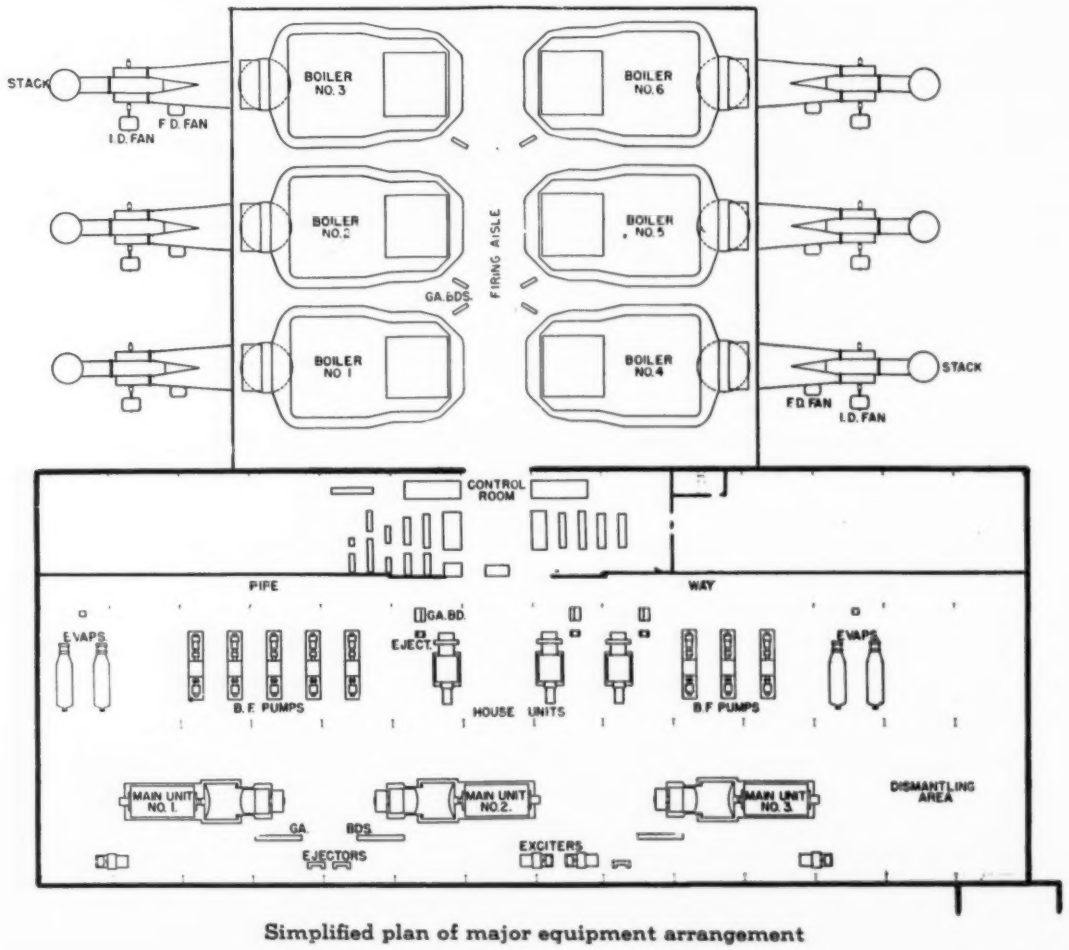
building rests on the equivalent of a 7-ft thick concrete mat. The intake tunnels, forebays, water-storage tanks and circulating-water discharge tunnels are all incorporated into one structure serving as the turbine pedestal foundation. Because of the site being in a region of earthquake faults, special attention was given to the seismic factor in laterally bracing all structures.

There are six stacks, one per boiler. These are self-supporting welded steel shells 11 ft O. D. and 200 ft high which were welded together at the plant site. The complete stack was then raised into position by means of gin poles. Reinforced gunite is used as a lining, which tapers in thickness from 8 in. at the base to 3 in. at the breeching opening, above which it remains constant to the top.

effected by means of a bypass damper. Although the nominal evaporation of each unit is 475,000 lb per hr at 1405 psi and 950 F at the superheater outlet and 420 F feedwater temperature to the economizer, the maximum guaranteed continuous output for a two-week period is 550,000 lb per hr.

Each unit is equipped with revolving and retractable soot blowers for the superheater section, revolving and stationary units for the economizer section and furnace wall boxes for possible future installation of additional blowers. Initially the soot blowers are being operated with saturated steam reduced to 600 psi.

Automatic pneumatic controls are installed to regulate fuel and air to the furnace, steam temperature, fuel oil temperature, air temperature out of the steam air heater,



Simplified plan of major equipment arrangement

Steam Generating Units and Auxiliaries

The six initial units are each of the three-drum design with plain tangent furnace wall tubes and three-stage superheater, fin-tube economizer, and Ljungstrom regenerative air preheater. Steam air heaters are also provided to preheat the air to the Ljungstrom heaters at light load. Firing is through the front wall by nine combination wide-range burners arranged in three tiers. Not only has space been provided for the future installation of coal bunkers, pulverizing mills and dust collectors, but the furnace volume and superheater tube spacing has been designed to accommodate solid fuel firing without affecting capacity due to a slower rate of combustion or slagging troubles. Steam temperature control is feedwater flow and feedwater pressure. Also, a purge interlock system is provided for automatic operation. Any or all of the boilers may be operated automatically, by direct hand control or remote hand control, as occasion dictates.

Units Nos. 4 and 5, now under construction, will each have a nominal rating of 850,000 lb per hr of primary steam at 1525 psi with 1000 F at the superheater outlet and reheat to 1000 F. They are of somewhat different design from that of the units now operating in that they will incorporate a single drum, a two-stage superheater, and employ both a bypass damper and a spray-type desuperheater for steam temperature control as required for the reheat cycle.

Fans

Mechanical draft on each of the present units is provided by one forced- and one induced-draft fan, located outdoors under ductwork between the boiler house and the stacks. The forced-draft fans have inlet-vane control on the double-inlet and louver dampers at the outlet of the scroll. They are of the high-speed type with backward curved blades. The induced-draft fans also have double inlets, but have louver dampers on both inlets and outlets. Provision has been made in the fan housings to accommodate larger wheels should there be a changeover to pulverized coal.

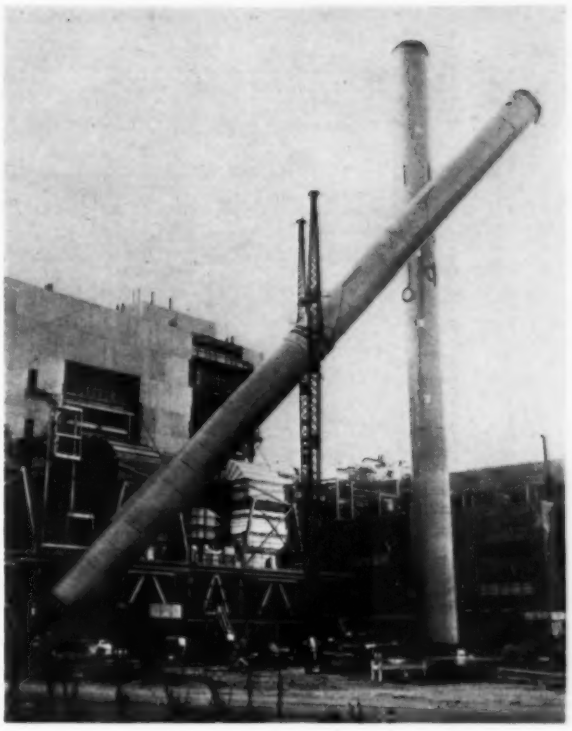
Pumps

Because of the very quick pickup feature, all the boiler feed pumps must be able to assume full load at a moment's notice. For this reason, as previously mentioned, pumps that are not carrying load are maintained in hot standby condition. The compiete pumping units are on a single baseplate with a common motor drive between the low- and high-pressure pumps. The piping arrangement is such that both the suction and the discharge nozzles extend vertically downward from the casing beneath the operating floor level. These pumps are located on the main operating floor adjacent to the turbine-generators. Two pumps are able to carry full load on a boiler unit.

Feedwater Heating

The closed feedwater heaters are of the horizontal U-tube type, with tubes in the lowest pressure heater of Admiralty metal and those of the other heaters of 80–20 cupro-nickel. The deaerating heaters, which function after No. 5 closed heater at 5 psig, are of the direct-contact tray type.

Incorporated in the heat cycle for Units 1 and 3 are two evaporators, arranged to give either single- or double-effect operation. They take steam from the third bleed point on the main turbines and discharge vapor to either their own condensers or to the main condensers. The evaporator condenser operates just ahead of No. 3 heater, as shown in the heat-cycle diagram.



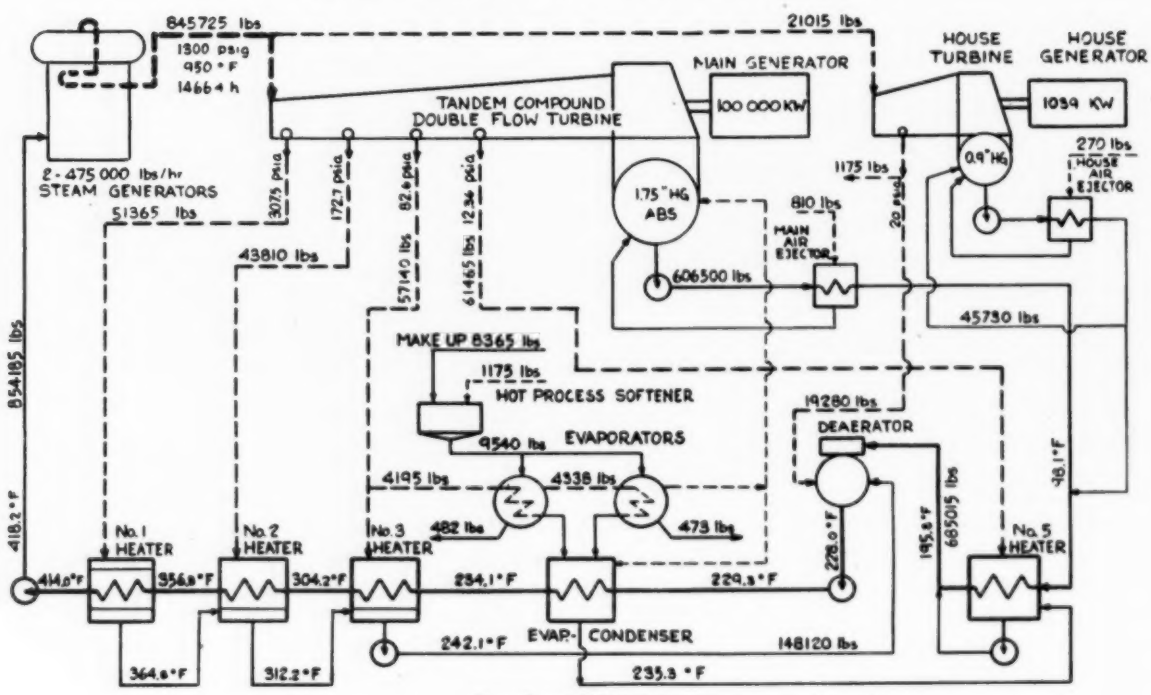
Raising stack for No. 2 boiler

Raw makeup water is pumped from the large storage reservoirs located under the main condensers and through the hot-process softening system before being introduced into the evaporators.

Turbine-Generators and Their Auxiliaries

Each of the three main turbine-generators has a nominal rating of 100,000 kw when operating at 3600 rpm, 1300 psig and 950 F at the throttle. They are of the tandem-compound, double-flow type with four extraction pressures of 308, 173, 83 and 12.4 psia, corresponding respectively to the 10th, 13th, 16th and 21st stages. An extraction point at the 19th stage is blanked off. The high-pressure element has 19 stages and there are 5 stages in the double-flow, low-pressure end. The

Two views of boiler-feed pumps—left, five Ingersoll-Rand pumps with main turbine-generators shown in background; right, three Byron Jackson pumps with evaporators in background



Heat cycle diagram

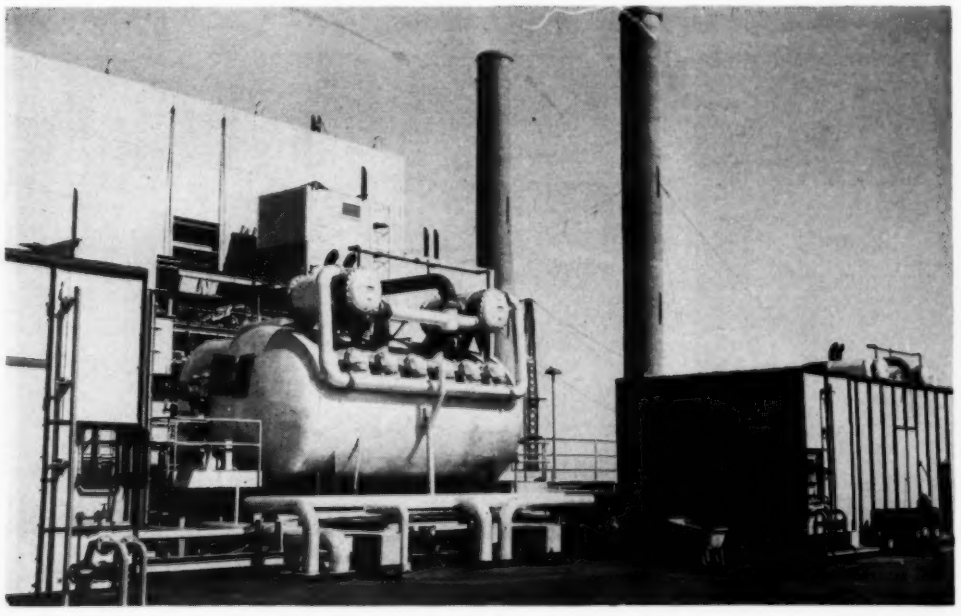
generator has a 129,375-kva rating at 15 psig hydrogen pressure and delivers 3-phase, 60-cycle current at 13,800 volts.

In addition to the main units there are three house turbine-generators which provide an independent source of energy for the electrically driven auxiliaries. These units are rated at 7500 kw, taking steam at 1300 psig, 950 F and exhausting at 4 in. Hg abs. The turbines are of the impulse type, have six stages and an extraction point at the third stage to supply 5-psig steam to the deaerators. Although designed for both condensing and noncondensing operation, the house turbines normally operate condensing.

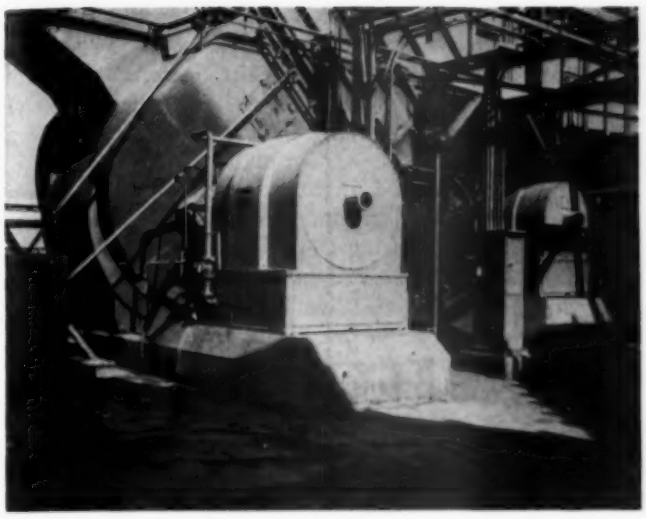
The condensers serving the main units each contain 50,000 sq ft of surface and are of single-pass design with water boxes divided vertically so that either half may be operated while the other is out of service for cleaning. The tubes are rolled at both ends into Muntz metal tube sheets. Cast-iron water boxes, steel-plate exhaust connections and an integral steel-plate hotwell complete the main assembly.

Although the exhaust neck is welded solidly to the turbine inlet flange, 230,000 lb of the weight is supported by eight springs under each of the four condenser feet. A welded-steel, bellows-type expansion joint at the outlet end takes up any unequal expansion between the shell and the tube bundle.

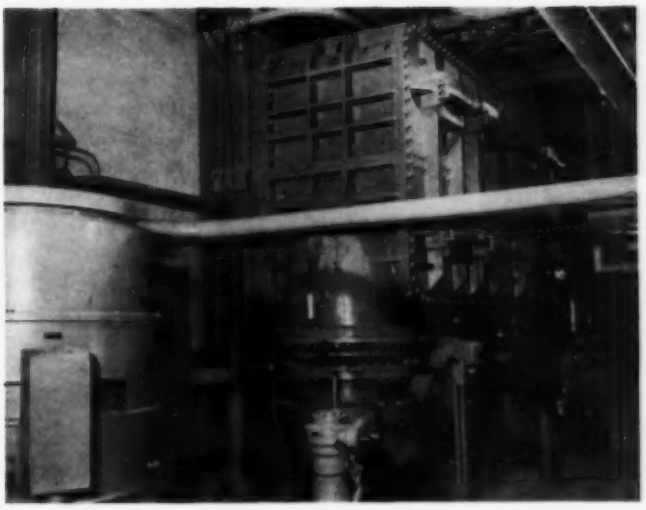
Each condenser is served by two vertical, single-stage,



Deaerating heater located on auxiliary bay roof



Forced- and induced-draft fan motors, induced-draft fan and ductwork in foreground



View of the main condenser as seen from the inlet end

mixed-flow circulating pumps. These are each rated at 44,800 gpm when operating against a total head of 21 ft. The casings and columns of these pumps are of cast iron and the impellers and shafts of stainless steel. Water-lubricated cutless rubber bearings and monel shaft sleeves are provided.

The 1600-gpm, five-stage condensate pumps are also of the vertical type.

Condensers for the house turbines are of the single-pass type containing 4000 sq ft of surface in aluminum-brass tubes. They have steel shells and cast-iron, divided water boxes. Although attached solidly to its turbine exhaust flange, the condenser has auxiliary spring supports to take care of expansion. Circulating water is furnished by the pumps serving the main condensers.

A chlorination system of the intermittent-dosage type is installed to combat marine accumulations on the tubes.

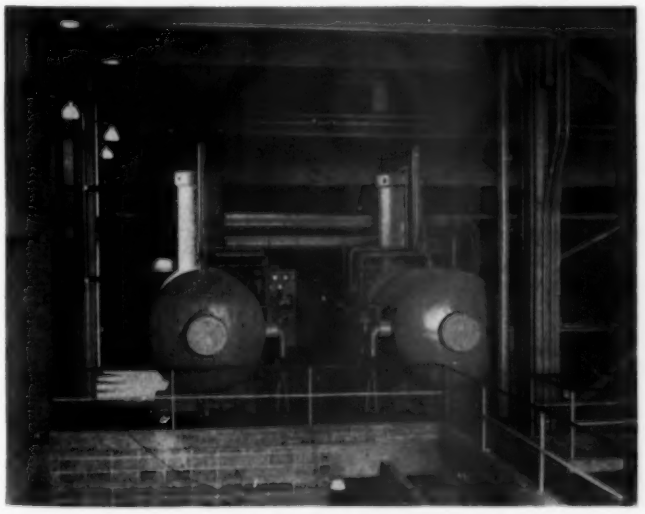
Some Electrical Features

Only some high spots of the electrical system will be touched upon here.

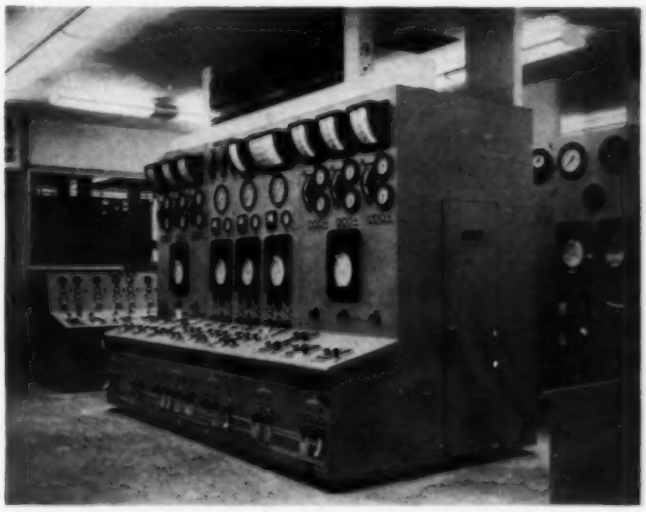
The main generators, which are rated at 129,375 kva at 80 per cent power factor when cooled by hydrogen at 15 psig, are of the single-winding type, wye-connected, with the neutral grounded through the primary of a 50-kva distribution transformer with resistance-loaded secondary. Connection to the main transformer of each machine is by a 13,800-volt isolated-phase bus capable of carrying the nominal generator current of 5410 amperes at full rating. There is no switching on the low side and the stepped-up voltage to outgoing transmission lines is 230,000 volts.

The principal auxiliaries operate at 4160 volts taken from two sources: namely, a 10,000-kva house transformer, connected to the main generator isolated phase bus, through an air circuit-breaker, and a house turbine-generator. These sources supply a 4160-volt ring bus for each unit. Lower voltage auxiliaries are supplied from this ring bus through two 1000-kva transformers which provide 480-volt power. Further step down to a 120/-208-volt four-wire system is by means of voltage-regulated transformers.

In the event of trouble on the transmission system, such as to jeopardize operation of the plant auxiliaries, automatic relays will disconnect the house transformers from their main generators and the house generators will carry the full auxiliary load.



A pair of evaporators



Master control board

A control room for the steam plant is located between the auxiliary bay and the firing aisle at the main operating level. This provides centralization of electrical and steam control boards. Here maximum visibility, through glass partitions and wide openings, is head of the firing aisle and the turbine and auxiliary bays. The openings into the firing aisle and the auxiliary bay are protected by automatically closed fire doors which would retard the entrance of smoke into the control room should a fire occur. This room contains fifteen boards with panels for each main unit and its associated auxiliaries, in addition to other boards for non-unitized equipment. The major equipment controls are located on bench-boards grouped around the operator's booth.

A second control point for the main generators is in the switchyard control building where most of the controls and the associated intruments are duplicated. However, the switchyard operator's function is to control only reactive power output, leaving the synchronizing and control of real power to the steam plant.

Engineering design and construction were carried out by the Bechtel Corporation, San Francisco, in cooperation with the Engineering Department of the Pacific Gas & Electric Company. For information upon which the foregoing article is based we are especially indebted to C. C. Whelchel, chief of the Steam Engineering Division of the Pacific Gas & Electric Company and J. N. Landis, chief power engineer of the Bechtel Corporation.

PARTIAL LIST OF MECHANICAL EQUIPMENT FOR FIRST THREE UNITS AT CONTRA COSTA STEAM PLANT OF PACIFIC GAS & ELECTRIC COMPANY, ANTIOCH, CALIFORNIA

Main Turbine-generators

Three 100,000-kw units (normal rating at 1300 psig, 950 F); tandem-compound, double-flow type with 19 high-pressure and 5 low-pressure stages; 5 extraction points (one blanked); 3600 rpm. Generator—3 phase, 60 cycles, 13,800 volts, hydrogen-cooled.—*General Electric Co.*

House Turbine-generators

Three 7500-kw (normal rating at 1300 psig, 950 F); 6-stage, impulse-type condensing with extraction at 5 psig. Generator—3 phase, 60 cycles, 4160 volts; air-cooled.—*General Electric Co.*

Main Condensers

Three single pass with divided water box, deaerating and reheating hotwell, 50,000 sq ft.—*Ingersoll-Rand Co.*

Main Circulating Pumps

Six vertical, single-stage, mixed-flow type; 44,800 gpm at 21 ft total head; driven at 435 rpm by 350-hp *Westinghouse* motors.—*Ingersoll-Rand Co.*

Main Condensate Pumps

Six vertical, 5-stage centrifugal type, 1600 gpm at 200 ft total head, driven by 125-hp *Westinghouse* motors.—*Ingersoll-Rand Co.*

Main Air Ejectors

Three twin-element, two-stage, steam-operated.—*Ingersoll-Rand Co.*

House Condensers

Three single-pass, 4000-sq ft, radial-flow, divided water box, deaerating hotwell.—*Westinghouse Electric Corp.*

House Condensate Pumps

Six horizontal, two-stage centrifugals, 180-gpm at 200 ft total head.—*Westinghouse Electric Corp.*

House Air Ejectors

Three twin-element, two-stage, steam-operated units.—*Westinghouse Electric Corp.*

Condenser Vacuum Pumps

Two "Hytor" type, 520 cfm at 15 in. Hg vacuum.—*Nash Engineering Co.*

Steam Generating Units

Six bent-tube, three-drum, radiant-type rated at 475,000 lb per hr continuous and 550,000 lb per hr for two weeks at 1405 psig, 950 F; *Elesco* three-stage superheaters; bypass damper control; *Elesco* finned-tube economizers (420 F inlet and 489 F outlet water temperature); Ljungstrom vertical-shaft air preheaters.—*Combustion Engineering-Superheater, Inc.*

Oil and Gas Burners

Nine wide-range, combination type per boiler.—*Peabody Engineering Corp.*

Soot Blowers—*Diamond Power Specialty Corp.*

Steam Air Heaters

Six Aerofin extended-surface type, 612,000 lb of air entering at 30 F.—*Buffalo Forge Co.*

Boiler Feed Pumps

Five units with primary and secondary pumps on common shaft. Primary pump—horizontal split casing, single suction, diffuser type, 580,000 lb per hr at 550 psi differential pressure. Secondary pump—horizontal, 8-stage, solid case, diffuser type, 550,000 lb per hr at 1470 psi differential pressure, driven by 2000-hp *G.E.* motor.—*Ingersoll-Rand Co.*

Three units with primary and secondary pumps on common shaft. Primary pump—three-stage, opposed impellers with double suction on first stage, horizontal split casing, 580,000 lb per hr at 550 psi differential pressure. Secondary pump—9-stage, opposed impellers with double suction on first stage, split inner volute casing and solid outer casing, 550,000 lb per hr at 1470-psi differential pressure; driven by 2000-hp *G.E.* motor.—*Byron Jackson Co.*

Feedwater Heaters

Three No. 1 closed, horizontal, U-tube, 4-pass, 400-psig shell design.—*Griscom-Russell Co.*

Three No. 2 closed, horizontal U-tube, 2-pass, 250-psig shell design.—*C. F. Braun & Co.*

Three No. 3 closed, horizontal, U-tube, 4-pass, 150-psig shell design.—*C. F. Braun & Co.*

Three No. 5 closed, horizontal, U-tube, 4-pass, 150-psig shell design.—*Griscom-Russell Co.*

Three deaerating heaters, direct-contact, tray-type with vent condensers.—*Cochrane Corp.*

Evaporators

Two double-effect, horizontal, 2-pass, 41,000 lb per hr; also two evaporator condensers.—*American Locomotive Co.*

Feedwater Treatment

One hot-process treating plant.—*Graver Water Conditioning Co.*

Forced-Draft Fans

Six with inlet-vane control and outlet dampers; 180,000 cfm at 100 F and 18 in. water; driven at 1187 rpm by 700-hp *Westinghouse* motor.—*American Blower Corp.*

Induced-Draft Fans

Six with inlet-louver dampers and outlet dampers; 315,000 cfm at 420 F and 10.1 in. water; driven at 498 rpm by 900-hp *Westinghouse* motor.—*American Blower Corp.*

Combustion and Feedwater Control

Automatic control of combustion, steam temperature, fuel oil temperature, air temperature leaving steam air heater, feedwater flow and pressure and pump regulation.—*Bailey Meter Co.*

Steam and Boiler Piping—*Midwest Piping & Supply Co.*

Fire Lines—*Grinnell Co.*

Expansion Joints—*Badger Mfg. Co., Chicago Metal Hose Corp., Fann Groves Co., Garlock Packing Co., Yarnall-Waring Co.*

High-Pressure Valves—*Chapman Valve Co., Crane Co., Wm. Powell Co.*

Blowoff Valves—*Yarnall-Waring Co.*

Insulation for Piping and Ducts—*Armstrong Cork Co.*

Engineers and Constructors—*Bechtel Corporation, San Francisco, California*

Space does not permit the listing of the numerous instruments, small pumps, tanks and other miscellaneous apparatus, nor electrical equipment, all of which has an important place in the makeup of a modern power station.

Trends in Application of Deaerating Heaters for Boiler Feedwater

By V. J. CALISE and R. K. STENARD

Graver Water Conditioning Company

Following is an abridgment of a paper presented at the 1951 ASME Fall Meeting, held at Minneapolis, explaining considerations in design details of deaerating heaters for industrial power plants and central stations. Attention is given to steam and water flow, types of vent condensers, tray versus spray-type heaters, materials of construction and water storage.

ORIGINALLY, boiler feedwater open or deaerating heaters were developed to effect substantial savings in fuel by recovering the heat in the steam discharged from engines or turbines. At present, proper deaeration of feedwater is as essential as removal of scale-forming constituents such as calcium, magnesium salts, silica and other materials.

In most boiler feedwater applications, corrosion and pitting of boilers and accessories are caused mainly by the presence of non-condensable gases such as oxygen and carbon dioxide or by low pH in the feedwater. Although pretreatment can increase the pH of the feedwater, oxygen and free carbon dioxide must be completely removed in a properly designed feedwater deaerating heater. Experience has shown that an oxygen concentration of not more than 0.005 ml per liter in feedwater, as measured by a precise, modern test method, is the upper limit for most high-pressure boiler plants to avoid corrosion difficulties.

An effective deaerating heater must be designed to do the following:

1. Heat the feedwater as high as possible.
2. Vigorously boil and scrub the heater water with fresh steam which can carry out to the liquid surface any remaining minute traces of oxygen or carbon dioxide.
3. Maintain as low as possible the partial pressure of oxygen and carbon dioxide in the steam atmosphere in contact with water; particularly at the point where deaerated water separates from the steam.
4. Continuously withdraw the oxygen and carbon dioxide (non-condensable gases) from the proper place in the heater.
5. Produce a deaerated effluent containing the guaranteed low quantities of oxygen and other gases regardless of fluctuations in load steam pressure or steam temperature in the heater or the amount of oxygen and other gases initially present in the feedwater.

Table I is a summary of some of the various heater designs now available with the many special features and modifications required by modern boiler plant operation.

Industrial Power Plants

PLANT I

Recently a large industrial manufacturer in the Midwest installed a 500,000-lb per hr two-stage lime-soda-phosphate hot-process-softening plant including two 500,000-lb per hr tray-type deaerating heaters to treat feedwater for 900-psig boilers and turbines. In the de-

TABLE I

Summary of Modern Deaerating Heater Design Characteristics

General Type Designation	Relative Flow of Steam and Water	Vent Condenser: External Tubular	Vent Condenser: Internal Hood	Incoming Water Distribution	Incoming First Stage Water Heating	Second Stage Deaeration	Relative Partial Pressure of Non-Condensible gases at point where steam and heater water separate		Steam Pressure Drop, psi	Water Pressure Drop, psi	
Tray Cross Flow Parallel Flow Counter Flow	a) Cross Counter b) Parallel c) Directly Counter	U Tube or Straight Tube	a) Self-adjusting spray valves (with spray valves these tray heaters are spray-tray type. b) Fixed orifices in Distrib. Pipe	a) V-notched weir box plus distribution pans & trays. b) Fixed orifices in pipes plus distribution pans & trays c) Spray valves & distribution pan	a) Heater Trays b) Spray Valves	Trays	Cross Parallel Counter	Low High Low	0.1 - 0.3	a) Weir trough Distributor b) Spray Valves c) Fixed Orifices	0.5 2.0 0.5-3.0
Spray	Countercurrent flow in two stage steps. Parallel flow in second stage.	"	"	a) Self adjusting spray valves b) Fixed orifices in pipe	spray valves	Scrubber or Reboiler		Low	0.3 - 0.6	2.0	
Atomizing	Countercurrent flow in two stage steps. Atomization in second stage.	"	"	"	spray valves	Steam jet atomization with fixed or variable area orifice		Low	0.5 - 2.0	2.0	

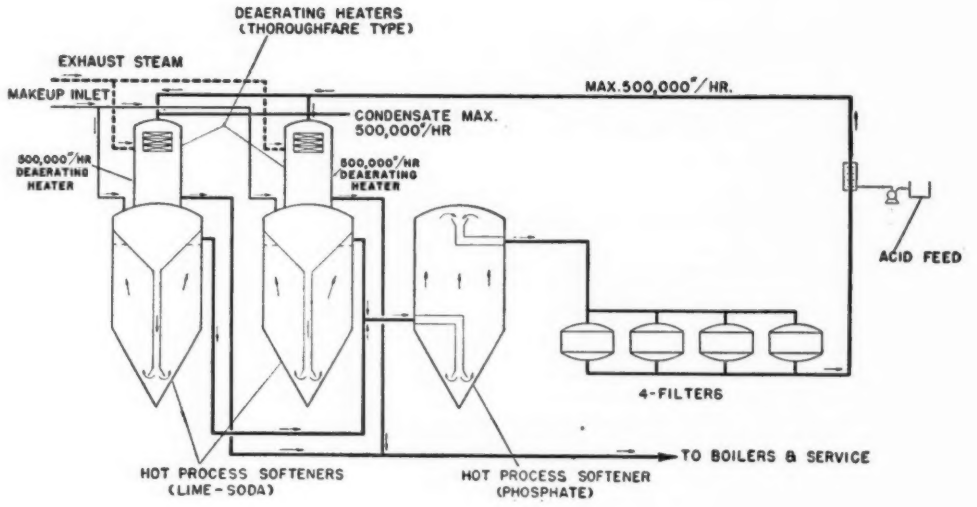


Fig. 1—Two 500,000-lb per hr deaerating heaters following hot-process softener

sign of the deaerating heaters for this installation, there were five main considerations:

1. Thoroughfare versus non-thoroughfare arrangement of steam and water flow relative to each other and hookup of hot-process softeners relative to deaerating heaters with location of vent.
2. Internal versus external tubular vent condensers.
3. Tray versus spray deaerating heaters.
4. Materials of construction.
5. Amount of deaerated water storage.

1. Steam and water flow

Fig. 1 shows a flow sheet of this installation. Fig. 2 shows a modified heat cycle diagram for this plant. The incoming cold lake water is sprayed through a rosette of spray valves into the steam space in the two main hot-process units and heated to within 2 deg F of steam operating temperature of 249 F and 15 psig. Lime-soda treated water from the first-stage units enters the second-stage phosphate units and passes through filters. It is then pumped through a corrosion-resistant mixer tube into which acid is fed with pH control to reduce pH down below 9.5 to avoid phosphate deposits in the stage heaters and economizers and also to reduce carbon dioxide in the steam. This makeup feedwater enters one of two vertical stainless steel-clad crossflow tray-type deaerating heaters employing three tiers of tray units. Hot return condensate up to 500,000 lb per hr is treated for oil removal and fed into the other deaerating heater, or it can be mixed with makeup and flow into both heaters uncontrolled. Exhaust or makeup steam reduced from 360 psig to 15 psig pressure enters the heaters, passes over the trays and flows to the hot-process softeners where non-condensable gases are vented by means of an internal-vent tube. This is the "thoroughfare" steam system employed on many combined hot process and deaerating heater installations.

The thoroughfare steam and water flow arrangement was selected for several reasons at this plant:

(*a*) The thoroughfare hookup with venting of heater vapor and gases through the hot-process softeners provides the maximum flow of steam through the heaters and therefore assures optimum deaeration with the most economical tray heater design.

(*b*) A stepwise hookup employing higher pressure 30–40 psig steam in the heaters and 15 psig in the hot-process softeners and with hot treated effluent entering the heaters and separate vents on the heaters and hot-process softeners would have provided less steam flow for deaeration. This would have necessitated another low-pressure reducing station and an additional low-pressure steam line which would have no other usage and also would have required a slightly larger tray section.

(*c*) Steam at 60 psig is frequently contaminated with oils and chemical products which would have presented a problem of contamination, and therefore was not usable.

(*d*) Since the plant was designed with duplicate first-stage hot-process softeners, duplicate deaerating heaters were purchased so that a combination of one first-stage hot-process tank and one deaerating heater can handle full load if necessary for any reason. However, with two units available, it was decided best to operate one heater on treated makeup and one heater on condensate only. By separating these two liquids, the net carbon dioxide content of the steam is reduced.

(*e*) Since the heater tanks were mounted over the hot-process softeners, there was no increase in cost of steam piping with thoroughfare hookup.

2. Internal vent hood versus external tubular vent condenser

The raw lake water used contains considerable amounts of calcium bicarbonate hardness. Accordingly, with the possibility of low load operation at times, as well as considerable variations in load, it was felt that an external

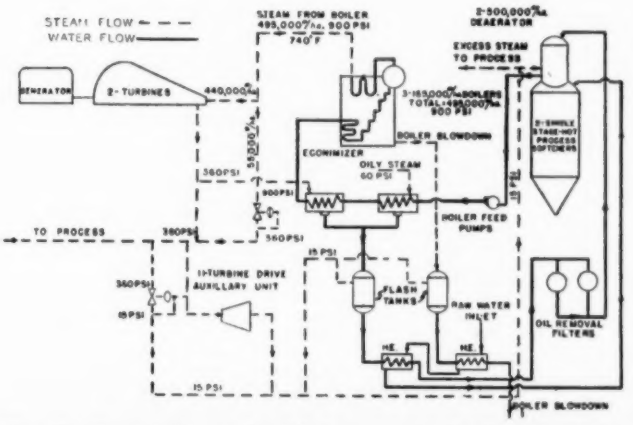


Fig. 2—Heat balance for Midwest manufacturing plant

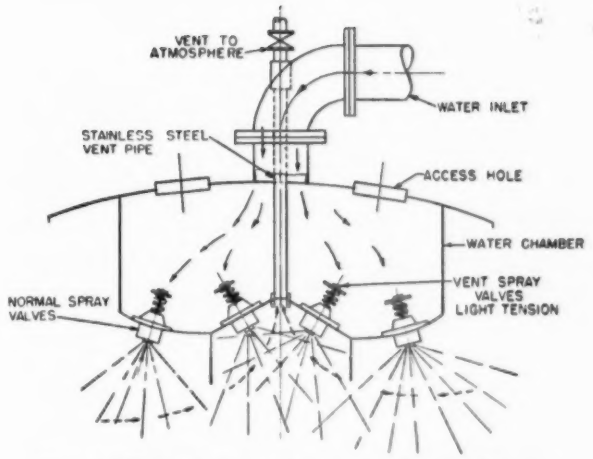


Fig. 3—Typical internal vent condenser design

tubular vent condenser would plug up too fast and require frequent maintenance. Accordingly, an internal vent in the hot-process tank was employed with the outlet for the non-condensable gases discharged from the hot-process tank located in a curtain of cold water spray. A typical design of this kind is shown in Fig. 3 and is employed successfuly on many installations where similar problems are encountered.

3. Tray versus spray-type heaters, and
4. Materials of construction

The water entering these heaters was acid-treated to prevent phosphate deposit in stage heaters and lines and to reduce carbon dioxide in the steam. This meant that a low pH, hot, corrosive water had to be treated in the deaerating heaters at times. Proper corrosion protection was absolutely necessary in these heaters. Accordingly, all stainless clad shells, with stainless steel internal construction, were finally agreed upon.

The engineering and operating personnel at this chemical plant have had extensive successful experience with tray-type heaters. With hot condensate at varying temperatures of 130–200 F, a tray-type heater was felt by plant operating engineers to be more suitable for this type of operation. For these reasons, two crossflow-type, tray heaters, as indicated in Fig. 4, were selected.

5. Amount of deaerated water storage

Although the hot-process softeners provide more than 30 min of storage of treated, partially deaerated water, it was felt that 6 min storage in each heater at 500,000 lb per hr load on each was necessary and was provided.

These deaerating heaters were checked about one year after operation was started and yielded an average of 0.003 ml per liter of oxygen on three samples.

Plant II

A Midwestern steel mill recently installed a 1,000,000-lb per hr, hot-process, hot-zeolite plant, including a two-stage, spray-type deaerating heater of the same capacity to treat feedwater for 650-psig boilers. Plant load fluctuates in maximum increments of 25 per cent at infrequent periods but each load variation occurs over a time interval of 5 to 10 min. Although the deaerating heater was designed to handle up to 1,000,000 lb per hr of makeup, normally makeup will be less than 750,000 lb per hr with 250,000 lb per hr of condensate returns from turbo-generators and boiler plant accessories.

1. Steam and water flow

As shown in Fig. 5, incoming cold water from city mains passes through an external tubular vent condenser with a maximum temperature rise of 20 deg F at maximum flow and is then sprayed through a rosette of spring-loaded, self-adjusting covered spray valves in the Sphericone hot-process unit. This heats the water to within 2 deg F of steam-operating temperature of 227 F and 5 psig and removes more than 95 per cent of the non-condensable gases in the water. The partially heated raw water is treated with dolomitic lime for partial softening and silica reduction to less than 2 ppm. Precipitates are separated by low, continually decreasing upward rising rates through a sludge bed to a minimum of 2.1 gpm per sq ft at the 1,000,000-lb per hr load. The effluent from the Sphericone hot-process softener contains less than 5 ppm turbidity and is pumped through a battery of zeolite softeners in which it is softened to zero hardness. This treated water then enters the 1,000,000-lb per hr heater.

The thoroughfare arrangement, employing clean steam at 5 psig, was selected because of more assurance of good heater operation under all load, water and steam conditions. This is shown as Scheme "A" in Fig. 6.

2. Internal versus external tubular vent condensers

Since operating load on this steel mill is reasonably constant 24 hours a day all year round, it was felt that scaling of external tubular vent condensers, due to very low load operations and nightly or weekend shutdowns, would be minimized. Accordingly, external tubular vent condensers were employed on both hot-process softener and deaerating heater. The vent valve on the vent condenser on the deaerating heater is normally closed with the thoroughfare system in operation and all non-condensable gases are vented from the softener.

3. Tray versus spray-type deaerating heaters

The water being deaerated is low in alkalinity, although high in pH, and contains less than 1 ppm of total hardness when entering the heater. The operating and design engineers had had considerable successful experi-

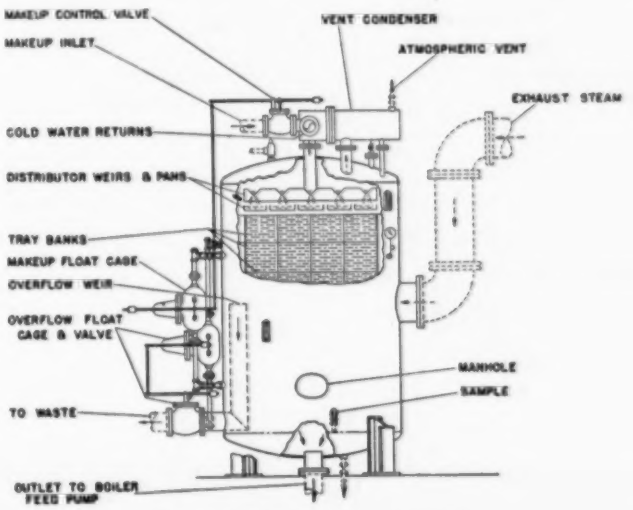


Fig. 4—Crossflow-tray heater

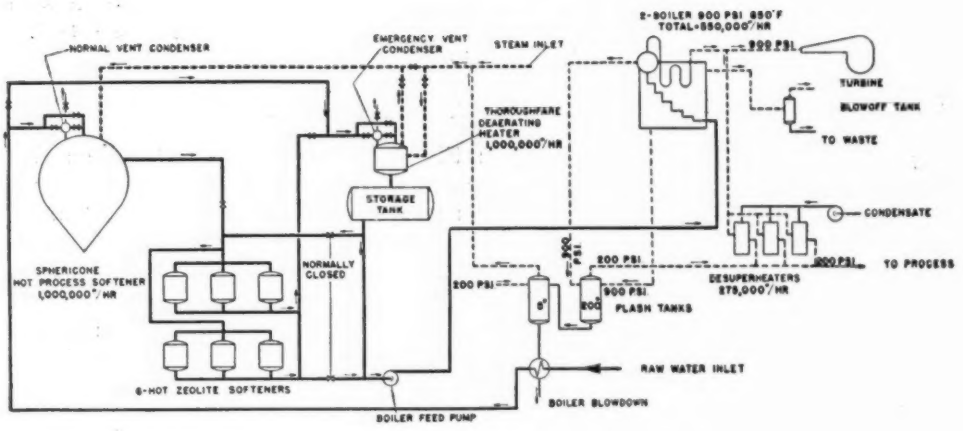


Fig. 5—Steel-mill installation of 1,000,000-lb per hr spray heater and softener

ence with spray-type deaerators. In addition, there were also savings in investment cost of approximately 10 per cent in favor of the spray-type heater. For these reasons, a two-stage, spray-type heater was selected.

4. *Materials of construction*

With this high pH, low-hardness partially deaerated water introduced into the steel water box and stainless steel covered spray valves of the deaerating heater, no protection of internal steel surfaces of the heater was furnished, except the usual stainless steel steam orifice plates and baffles. This is in accord with long experience with these treated waters in previous installations.

5. *Amount of storage*

Fig. 7 shows a cross-section of this 1,000,000-lb per hr heater. Approximately 10 min or 166,667 lb of deaerated water can be stored in this heater up to the lip of the overflow. Hot condensate up to 250,000 lb per hr can flow into the heater for deaeration. Additional storage of about 30 min or 500,000 lb is provided ahead of the heater in the hot-process-softener system.

Plant III

Several years ago, a large Eastern steel mill decided to expand its boiler plant and feedwater treatment facilities by adding equipment to the existing plant. Their water supply is taken from the Monongahela River and treated with cold lime soda in batch-type wood tanks, and in continuous cold-process softeners. Fig. 8 shows a heat-cycle diagram of this power plant.

With a scaling type of treated cold water such as this requiring deaeration, it was felt that no matter what type of deaerating method was employed in the second stage, i.e., trays or scrubber or atomization, the incoming water should be distributed in covered, self-adjusting, spring-loaded spray valves for optimum efficiency and maximum long-term operation.

Based on previous experience with this water, it was felt that the fixed orifices in distributor pipes, combined with heating and distribution trays, would scale up rapidly and require frequent maintenance. Furthermore, they are not as effective in first stage deaeration.

For second stage deaeration the operating engineers considered trays with relatively large tray areas more dependable than relatively smaller orifices that might clog easily and upset operation.

For these reasons two crossflow-design, two-stage, spray-tray deaerating heaters, each 500,000 lb per hr, and shown in Fig. 9, are mounted on a common storage tank. The first-stage, spray-valve rosette containing covered spray valves is flanged for easy removal. The two vertical heaters which operate at 5 psig are designed for 30 psig and are mounted on a large storage tank. Three tiers of weir edge tray units are employed. These provide positive weir spillage length when they overflow. Trays are of stainless steel. Heater shell is steel and spray valves are stainless steel.

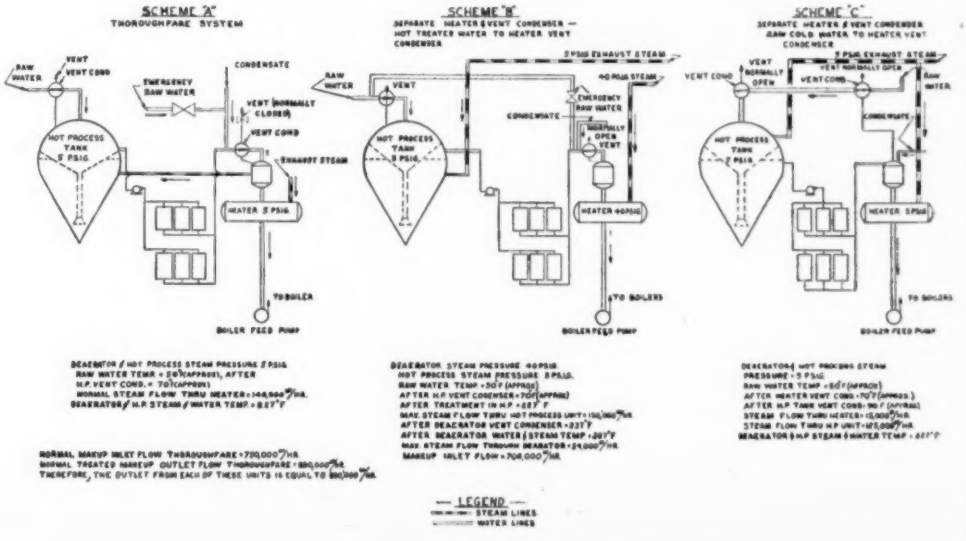


Fig 6—A study of three possible hookups of deareating heaters and hot-process softeners

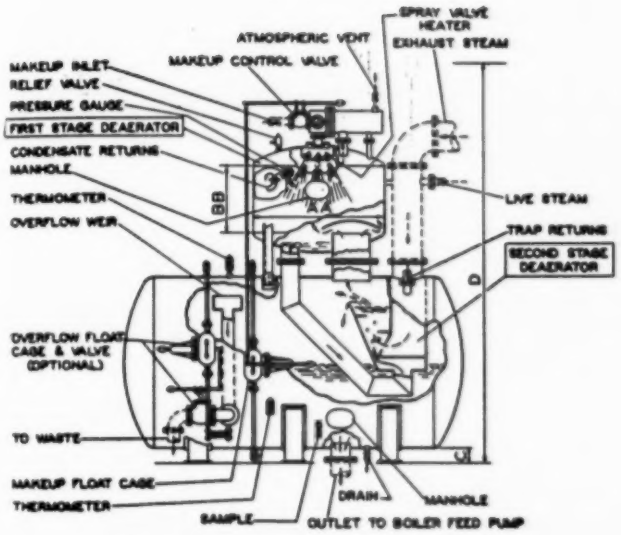


Fig. 7—Horizontal deaerating heater

The storage provided in these heaters is approximately 10 min to the lip of the overflow. Small amounts of condensate can also be deaerated in this heater. Flash preventers are employed to avoid steam binding of boiler feed pump when operating the heaters at higher pressures and any possible consequent falling off in pressure.

Connections of steam and water to these heaters are such that one heater can be inspected internally while the other is taking the load. Also spray valves can be inspected and replaced easily and trays can be treated with inhibited acid when and if scale is found present.

PLANT IV

An Eastern seaboard oil refinery has been operating a 450,000-lb per hr boiler plant including three fully automatic straight resin-sodium-zeolite units followed by a pH-controlled acid-feeding system.

The makeup entering the 450,000-lb per hr spray-type deaerator contains up to 90 ppm carbon dioxide. The deaerator is designed to deaerate up to 660 gpm of makeup at 60 F and up to 300 gpm of condensate at 150–210 F. Hot oily condensate is treated in preformed floc filters and then enters the heater.

In order to simplify the plant design the two-stage, spray-type deaerating heater was used as a degasifier and also as a deaerator. This two-stage spray heater was designed with a straight preheater mounted on a straight storage tank. An external tubular vent condenser was employed without fear of scaling because the influent is at low pH and has zero hardness from the zeolite softeners and low alkalinity after acid feeding. It contains cupro nickel tubes, Muntz metal tube sheets and a corrosion-resistant lining in the spray-valve water box. Stainless steel spray valves are employed. All other internal parts are of steel, except inlet steam distributor orifice in the second stage section, which is stainless steel. The water leaving the spray valves after carbon dioxide removal is normally at a pH above 8.5 and contains less than 0.2 ppm oxygen.

The inlet control valve is a double-ported valve having acid bronze body and trim to resist low pH makeup.

All portions of the piping and fittings after the acid-feeding system are corrosion-resistant.

Central Stations

Generally, central station deaerating heaters are operated at higher steam pressures than those for industrial plants. In addition, central station heaters usually employ steam extracted directly from one of the bleed stages on the turbine and their heat cycles are such as to require close control of deaerating heater steam operating pressure and temperature.

In many cases the heater manufacturer has been required to guarantee the operation of a deaerating heater with very small steam-pressure losses. The reason for this is that some power engineers charge the heater with an operating cost if the steam-pressure loss across it is excessive.

The higher steam-pressure loss results in a degradation of heat or lower water temperature and must be balanced by bleeding more steam from a higher extraction point in the turbine, thereby reducing the thermal efficiency of the turbine and the system.

Tray-type heaters can be operated with steam-pressure losses of less than 0.1 psig and with very small temperature differentials between available steam and incoming feedwater (which reduces amount of steam flow and velocity through the heater). This is accomplished at the expense of taller tray stacks, more trays or more tray spilling weir edge, and longer time of contact between water and steam in the heater. In general, the greater the steam flow and the higher the steam velocity through the heater, the simpler and less bulky the heater, regardless of type, and also the lower the cost of the heater.

Central station plants have generally favored tray-type heaters for the above reasons.

One of the main problems in central station operation

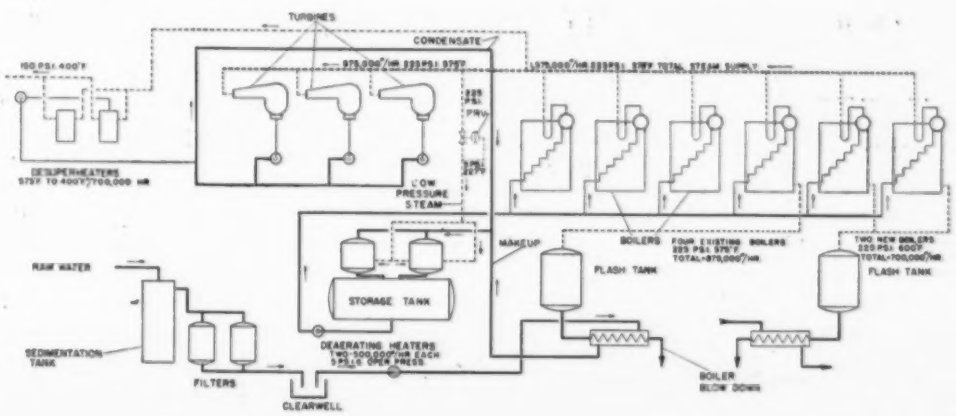


Fig. 8—Steel-mill installation employing two 500,000-lb per hr spray-tray heaters

of deaerating heaters is the problem of preserving good deaerator operation when turbine load falls, resulting in a drop in extracted steam pressure on the heater. In cases where this drop results in an operating pressure below atmospheric pressure, non-condensable gases must still be removed from the heater by proper venting. There are four methods employed for this purpose:

1. Vacuum equipment for non-condensable gas removal at sub-atmospheric pressure.
2. Next higher stage steam-pressure supply.
3. Constant lower pressure on heater regardless of load fluctuation.
4. Use of turbine condenser vacuum for discharging non-condensable gases.

METHOD (1)

One common method of operating deaerating heaters with turbine-extracted steam is to employ automatic ejecting equipment to create a vacuum and remove non-condensable gases from the heater when extracted steam pressure falls below a fixed point. When operating in this manner, the deaerating heater steam pressure will follow the bleed pressure from the turbine down as the load falls off.

Fig. 10 shows a simplified sketch of a typical vacuum cycle hookup employed with a 405,000-lb per hr deaerating heater to vent non-condensable gases. When steam pressure on the heater drops to 1 psig, a pressure switch is employed to initiate operation of a solenoid valve that opens a valve causing high-pressure steam to flow through a steam ejector. This creates a sufficient vacuum to suck steam and non-condensable gases from the vent condenser through the ejector and into an after-cooler, which is used to save the high-pressure heat. Drains from the after-cooler are usually discharged to waste because of high gas contamination. Gases are vented to atmosphere from the after-cooler.

METHOD (2)

Another method of operating the heater when turbine load and extraction pressure fall off is to use a pressure-reducing valve to supply steam to the heater from one of

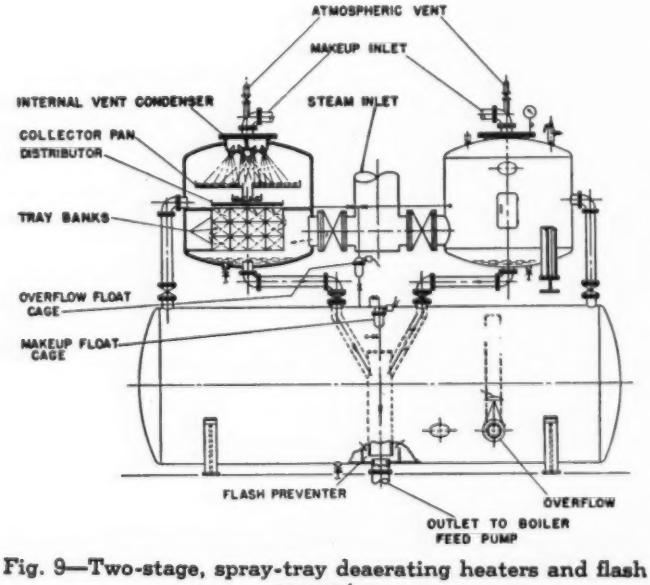


Fig. 9—Two-stage, spray-tray deaerating heaters and flash preventer

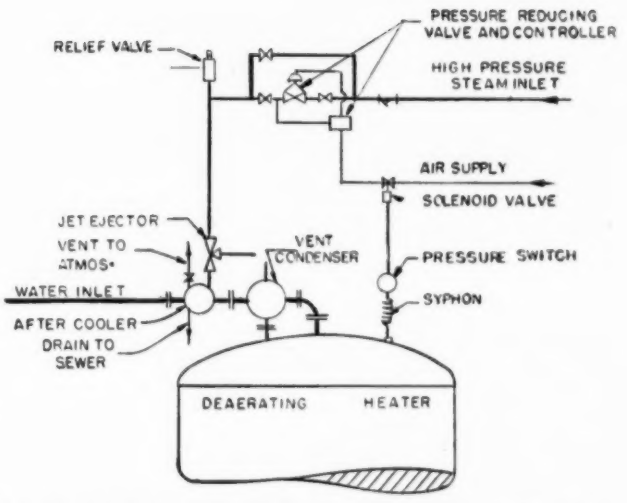


Fig. 10—Typical vacuum ejector arrangement to vent non-condensable gases

the higher bleed points on the turbine which is still above atmospheric pressure. Also, steam can be supplied from a higher pressure boiler steam header through a pressure-reducing valve. Normally, this supplementary pressure is maintained just above atmospheric at 5–10 psig at a constant value until bleed pressure builds up beyond this value due to load pickup.

METHOD (3)

Recently, a utility station in a Midwest city purchased three tray heaters, horizontal design, each to produce 350,000 lb per hr of deaerated effluent containing less than 0.005 ml per liter of oxygen. Influent condensate temperature is 174 F and effluent from the heaters is 233 F when employing 22 psia turbine bleed steam for deaeration purposes. Storage of deaerated water is approximately 2 min or 11,000 lb because of large surge storage provided in the system.

A constant steam pressure is maintained on these heaters by supplementary control of two other high-pressure steam sources, one from a higher bleed point on the turbine and the other from a high-pressure steam header.

METHOD (4)

In some cases where deaerating heaters are operated on steam extracted from turbines, plant practice is to operate the heater with an alarm actuated by a pressure switch on the heater. When the steam pressure on the deaerator falls to 1 or 2 psig due to a load drop on the turbine, the alarm is tripped and an operator will manually or semi-automatically operate a valve connecting the vent pipe from the heater to the turbine condenser which is at 1-in. or 2-in. Hg pressure. As an example of this, recently a 250,000-lb per hr tray-type deaerating heater was installed in a large Western central station. This heater provides a 10-min storage of deaerated water.

Normally the load on this deaerator will be constant night and day and seasonally because it services a boiler-turbine combination that operates at good efficiency.

Under these conditions, drop-off load periods will be very infrequent and a simple alarm device on the deaerator to indicate low pressure to the operators has been provided. This method of operation at low loads is not usually employed in larger stations, where load drops may be more frequent.

What Power Industry Designers Expect of Boiler Instrumentation*

By H. C. MITTENDORF
Combustion Engineering-Superheater, Inc.

It is urged that instrumentation be considered while the boiler design is in progress and that special attention be given to the location and number of furnace inserts. Various controls are discussed and the progressive development of board types is illustrated.

THE owners, their engineering men, both supervisory and operating, and their consultants decide questions which determine the basic design of a boiler unit. Also, within the manufacturer's organization are those engaged in the design of pressure elements; those who calculate the performance of these elements; others who design the burners, pulverizing mills, air heaters; still others who coordinate these items into a unit to fit the available building space. Touching all of these men are individuals who are expected to see that the designs can be fitted with instruments and to insure that connections are made in a way acceptable to the instrument manufacturers whose equipment is to be used. While most of these engineers are concerned only remotely with instrumentation, all are vitally interested in the results obtained from that instrumentation.

From the foregoing it will be apparent that a "boiler designer" is a multiplicity of engineering specialists whose needs and expectations in the instrument field range from nothing to the maximum. It is most important to think of instrumentation while the design is in progress. Starting with furnace pressure inserts, their location is a joint obligation of the boiler and instrument manufacturer. Most boiler furnaces are intended to be operated at approximately 0.15 in. w.g. draft at the apex. The elevation of the inserts in the furnace of a low set boiler is not very serious, but with a high furnace the draft difference between the bottom and the apex, due to stack effect, may be as much as $^3/_4$ in. No one, seriously, would think of applying the furnace draft connection at the furnace bottom, yet frequently drawings are received marked by consultants and manufacturers, to show the insert approximately half-way down. At such elevation a furnace draft controller set for a constant draft would cause operation at too high a draft for some loads and too little draft at others. Excessive draft produces too much air, not under control, at one rating and the possibility of a hot roof at other ratings. The company with which the author is associated has a standard calling for the furnace draft gage and controller to be connected to inserts located approximately 3 ft below the roof and 2 ft forward of the furnace exit tubes. Changes from this location often are necessary, but the standard serves as a guide.

Furnace inserts in water-cooled furnaces usually require the bending of tubes. Our practice is to use separate inserts for the gage and for the controller, thereby following the manufacturer's recommendations. Usually a separate test insert also is supplied. Unless these are aligned vertically and quite close together as many as six tube bends, sometimes in two planes are required. Although, we abide by the manufacturer's desires not to parallel these instrument connections, we believe benefit would accrue if one insert only were used, for then, regular cleaning would be imperative. Facilities for the test group to connect in their manometers, if they desire, should be possible by valving-off and the use of short nipples in the operating instrument tubing. This should be possible without disturbing the tubing to those instruments.

Number of Inserts and Taps

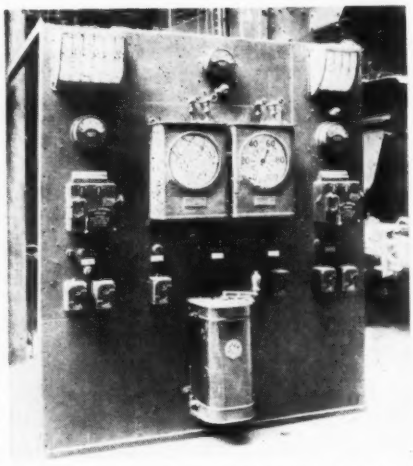
The use of one insert would save the boiler manufacturers not only material, but considerable drafting detail for the tube bending, shop and field labor, and field maintenance later. Where allowance must be made for differential expansion, the sealing often presents a design problem not easily met.

That which has been said in favor of one multiple tap connection for the furnace applies also to taps for air flow measurement, air flow control and superheat air flow anticipation. These taps, whether made on the gas side or in the air ducts, are not so expensive as those for the furnace because tube bending usually is unnecessary; but the desirability of one tap still applies.

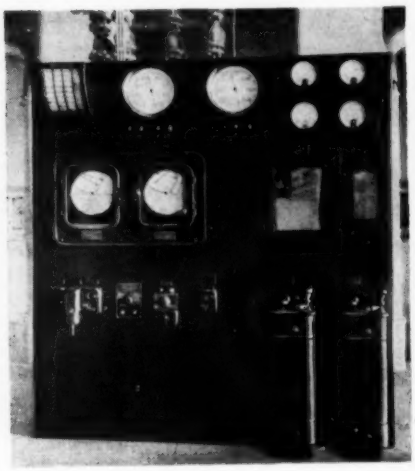
It has been customary to match air flow in the combustion control system against steam flow or to match fuel delivery to the air flow. In either case the differential draft or pressure loss has approximated 2 in. w.g. at the maximum flow. This somewhat high value is desired so that reasonable differentials exist at one-fourth to one-fifth of the maximum flow. At this reduced rating only about 0.10 in. is available.

In older boiler designs it was not a major problem to find some location where this high draft differential could be obtained. Frequently it was taken across convection gas passages. Often now, this possibility does not exist. Division of gas flow by damper movements, reduction of conventional boiler convection surface, and the use of bypass lanes have made it difficult to find

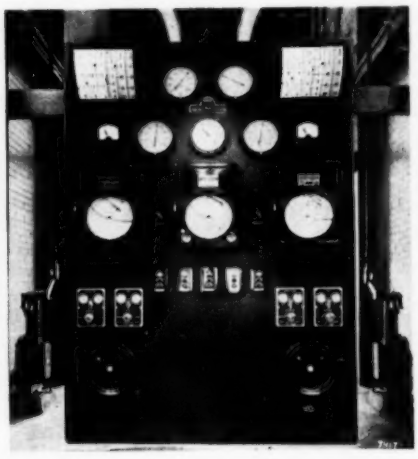
* From a paper presented at the Philadelphia Section Conference of the Instrument Society of America, a report of which appears elsewhere in this issue.



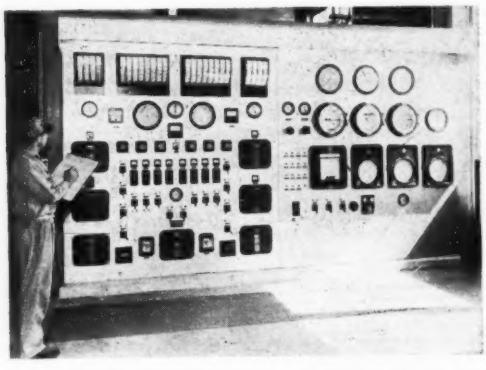
A metal board on which the boiler meter with pens for steam flow and air flow appears prominently



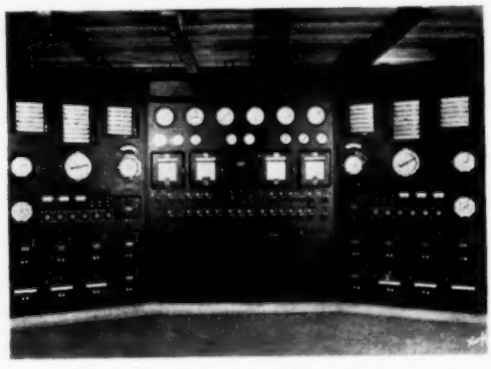
First of the flush instrument case designs, although not all equipment, is designed for flush mounting



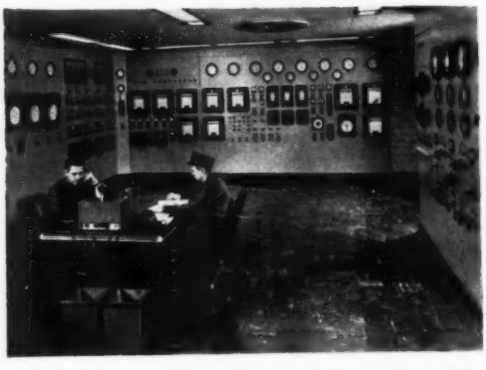
Forerunner of present board with most equipment flush mounted. Regulators for dampers at each side



A cubicle board with rear under air pressure to prevent pulverized fuel dust entrance. Operating instruments on one section with recorders on the other

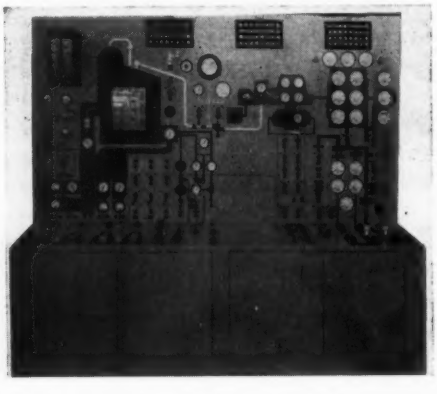


A two-boiler board with side panels set at an angle to center section, carrying master steam pressure controller



Half of centralized control room with operators' location air conditioned

A two-unit control room



Left—Front of a graphic board, containing no recorders, merely indicators and controllers with an outline of the unit, water and steam lines and generator circuits

a satisfactory location on the gas side. Resort has often been made to taking the measurement on the air side. This is satisfactory for oil- and gas-fired installations even though some arbitrary restriction may have to be imposed to obtain the required differential pressure. In pulverized coal fired installations it has the disadvantage of not measuring the mill tempering air. This can introduce, in extreme cases, a sizable air-flow error.

An admirable case for air-flow measurement in the air ducts is made by I. G. McChesney of the Rochester Gas and Electric Company in the August 1951 issue of COMBUSTION. His "air meter" has a nearly zero pressure loss.

It has been found that chemical type CO_2 instruments are as accurate as the Orsat when used for checking, but the time lag between sampling and analyzing makes them of little use when rapid load changes are occurring. But, even if the lag could be reduced appreciably, the spot sampling of flue gases always makes their readings questionable. Of course, this objection applies to all devices that draw samples from one or two locations. If stratification exists, a representative sample is not obtained.

Water level gage glasses usually are not classified as instruments, but they do present some difficulties on high-pressure applications. Mica discoloration and glass etching may be found after only a few weeks' operation. In part, this is the user's responsibility, caused by operating with too high and dirty solids concentration. It may be the designer's responsibility too, for he may have made drum connections where a nearly static water condition exist. However, when mica stands up no longer than six weeks it is indicative of the need for a more lasting material. Perhaps some of the newer plastics should be studied to protect the glass. Along with this study should go, too, a study for another method of connection for the present glasses. The author has been recently told of an improvement already effected by a rather simple connection change.

Water temperature measurements by thermocouples or resistance thermometers in wells have been satisfactory. Steam temperature measurements have been less so, although, in part, this is not the instrument's fault. Lag is the principal difficulty. The lag is largely in the well, which may delay the measurement a full minute. This well lag, added to the resistance, the capacity and the transportation lags makes the steam temperature control difficult. The anticipating means used to overcome the system lag need definite improvement.

The comments previously made about connecting operating and test instruments in parallel apply to thermometric measurements with minor revision. By this is meant that separate thermometer wells in water lines and steam lines for test purposes should be unnecessary. Instrument manufacturer's designs should provide for a transfer switch to enable the thermocouple transfer to be made without disconnection. Although, a separate connection in a gas duct is not an expensive item, high-pressure and temperature wells with labor to weld them in can become a sizable item.

Two-element feedwater controllers are a requisite for the smaller modern boilers, and the three-element design is desirable for large ones. Even with quite steady loads, single-element equipment on small boilers is likely to be a failure.

Combustion control is a subject which ranges from a simple pressure regulator controlling in parallel fuel, air and uptake damper, to the complex installations controlling two or more fuels fired singly or in-varying proportions.

The connection of combustion controller elements for stoker-fired furnaces and boilers has produced sharp disagreement between stoker engineers and the controller manufacturers. In general, on single- and multiple-retort machines our thoughts favor the furnace-draft controller connected to the uptake draft regulator, rather than to the forced-draft equipment. It is necessary especially when picking up load, or when the fire has become ragged, to view it frequently. This opening of inspection doors causes an inrush of air at that moment and results in reduction of air through the fuel bed if the forced-draft equipment is connected to the furnace.

Although the idea of regulating compartment air pressures automatically on traveling grate stokers was thought to have been discarded long ago, occasionally specifications call for it. The author knows of no successful installation.

While metering type controls are desirable, it is appreciated that many small units cannot stand the higher first cost and the owners lack the personnel to understand and service the metering designs. For such cases, a positioning system can be the only one justified.

From the viewpoint of the boiler operator, functional arrangement should take precedence in the location of equipment on the control board. Although a balanced design, meaning instruments with identical cases in neat vertical and horizontal rows, may look well on the drawing board and in a photograph, such an arrangement should be discarded if it does not bring together those instruments and controls which the operator must observe at the same time he is to discharge his duties properly.

Control board room general illumination is necessary but should not be too intense at eye level. Instrument illumination preferably should be greater than that normally now used. Each instrument should have its own lighting and some degree of adjustment to accommodate illumination to individual desires.

Tinted glass in instrument cases would help to reduce glare and reflections. Plastics might be better still.

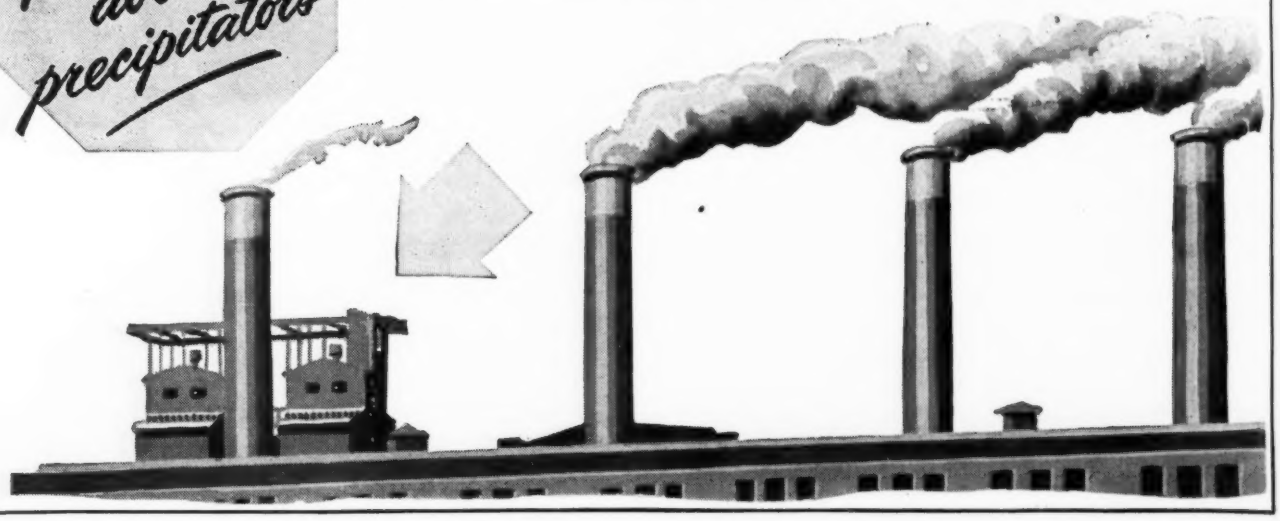
There is a trend in control board design to remove recorders from the direct view of the operator and to locate them on a remote panel. The idea is to shorten the control board by using indicators only. This puts it up to the operator to remember the trends of all the indicators. The author would like to see control board lengths reduced but not by removal of recorders. One way to shorten a board would be to get more records on one chart.

Such a chart would have an added advantage. At present, in order to study the charts, one must orient them for time, read some charts clockwise, others counterclockwise, some with pens on the left, others on the right; and when they have been turned and studied, there still is a question if the observed time of one agrees with the observed time of another. It is a clumsy operation to look at six records in this way.

Development of instruments and their mountings is portrayed interestingly in the accompanying views of control boards. The pictures themselves are self-explanatory.

Let's be practical about precipitators

This picture, based on an actual installation, shows the effectiveness of Koppers-Elex electrostatic precipitators. The stack at the left is served by two Koppers-Elex precipitators; the others are not so equipped. Because the stack discharge from the Koppers-Elex equipped stack is so clean this customer has ordered another Koppers fly ash precipitator.



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ASME Annual Meeting at Atlantic City

AN ATTENDANCE of over forty-six hundred marked the 1951 Annual Meeting of the American Society of Mechanical Engineers at the Chalfonte-Haddon Hall, Atlantic City, November 26–30. Papers on power and related topics occupied a major part of the program. Outstanding among these were nine dealing with central station experiences and performance with reheat installations which served as a sequel to a 1948 symposium on the subject dealing largely with design and economic factors. Other important sessions were devoted to spreader stoker performance at light load, feedwater problems, a series of three papers on the American Gas and Electric System, furnace performance factors, gas turbines, etc.

Retiring president J. Calvin Brown (left) greets the new president, R. J. S. Pigott

Among the non-technical events the Roy V. Wright Lecture deserves special mention. This was delivered by **W. C. Mullendore,** president of the Southern California Edison Company who, in a striking address, sounded a warning against the actions of a small but powerfully placed minority of international "do-gooders" who are recklessly over-committing this country to bear the burdens of the world. Declaring that after the hundreds of thousands of our young men had been sacrificed in winning World War II, we had permitted ourselves subsequently to be maneuvered into the present mess, he called for placing the freedom of the American citizen and conservation of the vital strength of our country above the claims arising out of the age-old quarrels and decadent policies of foreign nations.

A somewhat contrasting view was presented by **W. L. Batt** as the principal speaker at the banquet. As Minister in charge of the ECA mission in England, Mr. Batt reviewed what is now being done toward extending technical aid under the Point 4 Program and quoted from the reports of a number of British teams that had been sent to this country to study our industrial production methods. These, he believed, had been most helpful in explaining American spirit and accomplishment.

Among the several honors conferred at the meeting were the John Fritz Medal to Ervin G. Bailey for "outstanding engineering achievements in the field of combustion and distinguished service to his fellows in advancing the engineering profession"; the Hoover Medal to W. L. Batt in recognition of "leadership in engineering, management and public responsibility"; the ASME Medal to Glenn B. Warren for "leadership in the science and art of turbine design"; and the Richards Memorial Award to J. K. Salisbury for "outstanding achievement in mechanical engineering within 20 to 30 years after graduation."

Central Station Reheat Experience

In a paper on "Reheat Experiences at Port Washington" **M. K. Drewry** of the Wisconsin Electric Power Company took into consideration starting schedules after shutdowns of various durations, reheat economics, centralized control of auxiliaries, reheat temperature control, turbine overspeed control and availability of reheat units.

Since 1940, starting time following shutdowns of less than 30 hr has been 30 min, while rated load has been reached in $7\frac{1}{2}$ to 15 hr when starting from cold. The primary guide in turbine starting rate is axial expansion of the turbine cylinder, and experience has shown that this method affords a practical control even though it is not a perfect index of thermal stresses. Starting of the reheat turbines is essentially like that for straight condensing units. Piping and turbine are warmed together, there being no bypasses for pipe-warming purposes.

Thermal gains of 5.22 per cent over non-reheat rates have actually been realized on the basis of plant heat-rate records from 1935 to 1950. Reheater maintenance has been negligible, averaging $554 per boiler per year for units 1 to 4 during the years 1946 through 1950. This is less than $\frac{1}{2}$ per cent of the annual coal saving creditable to reheat. Total maintenance for the plant for 1950 was only 4.3 per cent of the fuel cost. Net dollar savings due to reheat have proved to be 4.05 per cent of annual fuel cost, and the extra investment for reheat (estimated to be $1.30 per kw) is repaid by reduced operating costs, at 60 per cent load factor, in $1\frac{3}{4}$ years. The five Port Washington reheat units are currently netting an annual overall saving (in operating expenses minus fixed charges) of $360,000.

By centralizing controls for boiler feed pumps, condenser auxiliaries and pulverizing mills at the boiler feed pump position, it is possible for one operator to perform functions formerly requiring three men. With other adjustments and reassignments a net reduction of 34 men in the station will take place, making the year-round personnel average equal to 0.57 employee per 1000 kw of rated capacity.

All units employ combination radiant-convection superheaters and all-radiant reheaters The combination superheater assures essentially constant steam

temperature throughout the load range, but the reheat temperature drops off at low loads. Improvements in the latter condition are expected at a new station at which the reheater and radiant superheater positions will be exchanged so that differential firing will transfer more heat at low loads into the reheater and less into the waterwalls.

While there have been no experiences with excessive overspeed due to reheat steam, the subject has been studied and certain precautions are taken. Downstream intercept valves are partially closed manually each shift to insure freedom of packings, stem bushings, and power pistons.

Experience amounting to 25.7 unit years shows that reheat does not impair reliability. Availabilities over this period were as follows: boilers, 95.2 per cent; turbines, 93.8 per cent; boiler-turbine units, 92.3 per cent. During the 1947–1950 period availability of the boiler-turbine units has averaged 94.6 per cent, and their forced outages have been 0.7 per cent. Over 50 per cent of the forced-outage time has been caused by generator trouble.

"Twenty-five Years of Reheat Operating Experience on the A. G. & E. System" was the subject of a paper by **S. N. Fiala** of the American Gas & Electric Service Corp. who stated that the successful and satisfactory operation of reheat units verifies past decisions and attests to the desirability of making future expansions along similar lines.

Beginning with a 40,000-kw turbine-generator supplied with steam by four chain-grate stoker-fired boilers which went into service at the Philo Plant of The Ohio Power Company in the fall of 1924, the A. G. & E. organization has made systematic progress. In the first installation which operated with 600-psig, 710-F primary steam and 710-F reheat steam controlled by a special boiler, the relation between kilowatts and number of boilers was roughly 10,000 kw per boiler. The 165,000-kw triple-compound unit placed in service at Philo in 1929 operated under similar steam conditions, but each of the eight boilers produced the equivalent of 20,000 kw. In 1930 at Deepwater Plant, progress was represented by an increase to 27,000 kw per boiler, operating at 1200 psig and fired with pulverized coal. A single boiler was designed in 1940 for Twin Branch to generate primary steam at 2300 psig, 940 F, reheating exhaust steam to 900 F and developing a total output of 75,500 kw.

Starting in 1947 a single boiler-turbine unit having a gross generation of 150,000 kw and steam conditions of 2000 psig, 1050/1000 F was projected. Five of these units are now in service, including three at the Philip Sporn plant, and two additional units are under construction. At the present time four still larger units are being planned, each of the single boiler-turbine type to generate 217,000 kw at 2000 psig, 1050/1050 F. Heat input to the boiler at full load will be 1.86×10^9 Btu per hour with an anticipated heat rate of just over 9000 Btu per net kwhr.

With one minor exception the procedure to be performed prior to initial start-up of a reheat unit is the same as required for a straight-through installation. Each element of reheater tubing is blown out with compressed air, and the connected reheat piping is cleaned out manually. Compared to the 18 hours required to start up the Twin Branch 2300-psig unit, the 150,000-kw boiler-turbines have been brought up to rating in $7\frac{1}{2}$ hours. Shutdown procedures have also been altered, and the period of cooling has been reduced from an average of 16 to 6 hours.

Regarding temperature control, experience has established the validity of controlling primary steam temperature by attemperation with feedwater. No superheater tubes have been lost, nor has there been an increase in turbine blade deposit. However, with this type of control wide fluctuations in temperature are possible unless attemperators are properly engineered, including foolproof instrumentation and interlocking.

Flexibility of operation is affected to some extent by the additional problem of reheat temperature control. Assuming a sound basic arrangement of superheater and reheater, the degree of flexibility depends on the quality of selection and application of control equipment. The boiler unit should not be used, except under unusual circumstances, as a heat accumulator to counteract insensitivities and malfunctionings of boiler controls.

Experience has indicated that reheat has no known effect upon chemical control of steam condensate, feedwater and boiler water as compared to straight condensing units. It is believed that reheat assists in reducing water-insoluble deposits in low-temperature turbine stages. Regarding overall availability and maintenance costs, the reheat cycle has negligible effect There is some reduction in turbine maintenance cost because of the lower moisture content of exhaust steam, but this is balanced by the increase in attention required by shutoff valves at the reheat turbine inlet. Problems brought about by high steam temperature have been compensated for by profits in efficiency gained.

H. E. Stickle of the Boston Edison Company presented a paper on "Operating Experience with Reheat at Edgar Station" where the reheat cycle has been in use since 1925. The first installation consisted of a single boiler with a topping turbine exhausting into 350-psig condensing turbines. The reheating of the topping-turbine exhaust, which was but a few degrees above saturation, was accomplished by reheater surface in the high-pressure boiler. In 1929 a second installation was completed, consisting of an additional 350-psig, 725-F condensing turbine, two 1200-psig, 700-F topping turbines and four 1400-psig, 750/750-F reheat boilers. There were cross connections at the 1200-psig level, the topping-turbine exhaust level, and at the 350-psig main steam header. The third addition was started up in August 1949 and incorporated a single boiler-turbine-generator unit rated at 81,250 kw with steam conditions of 1450 psig, 1000/1000 F.

From an operating engineer's standpoint, the inclusion of a reheater in a boiler results in only one more temperature to watch and to do something about when and if required. Assuming that the design engineer has provided suitable means for indicating and controlling the temperature, the incremental complexity due to reheat is practically nil.

In order to realize the savings in heat rate made possible by the reheat cycle, the unit must be available to run a high percentage of the load hours of the year. Mr. Stickle pointed out that the increased reliability that is being built into present-day power plant equipment is one of the reasons for increased interest in the more efficienct power cycles.

The third addition to Edgar Station, mentioned previously as having a rating of 81,250 kw, has demonstrated that the temperature of both the 1450-psig primary steam and the low-pressure reheated steam can be maintained at the design value of 1000 F at full load when oil is burned with a minimum of excess air. Regulation of the reheat temperature is by the burner tilt and that of the primary superheated steam by a desuperheater, the water for which has been taken from the deaerating heater. However, because of packing troubles with the reciprocating pumps, changes are being made to obtain the desuperheating water from the boiler feed line.

Such outages as were required on this unit were of a nature that could be scheduled for periods of low load, the principle cause of outage being fouling of the air heater due to low exit gas temperature and the high dewpoint of the combustion gases. However, since the first of the year a new method of cleaning has replaced washing and has been employed to maintain the air heater free for extended periods. This consists of burning coal for an hour each week, followed immediately by blowing with steam by an oscillating blower. This hour of coal firing apparently supplies a coating of sufficient ash for scouring action when the steam blowing is applied.

Performance figures for this unit during 1950 showed a net Btu per kwhr of 10,250, with 88.87 per cent load factor and 92.08 per cent availability.

The author concluded with the statement that over 25 years' use of reheat at Edgar Station has shown no operating difficulties that could be attributed to the reheat cycle and that, with the increased reliability now being built into plant equipment, the expected return on the additional investment should be obtained.

In a paper on "The First Year's Operation of the Dunkirk Steam Station" **J. N. Ewart** of the Niagara Mohawk Power Corp. stated that the performance of reheat units has been generally satisfactory even though there were some troubles of the type likely to occur in breaking in a new plant incorporating several untried developments. Operation of the two single boiler-turbine-generator reheat units[1] rated at 80,000 kw with turbine inlet conditions of 1450 psig, 1000/1000 F has introduced no new limitation in flexibility nor increase in operating personnel over non-reheat units.

Unit No. 1 was placed in commercial service on November 19, 1950, and Unit No. 2 followed on December 28. Up to October 1 of this year there were six unscheduled outages, equally divided between the boilers and turbines. Monthly net station heat rates ranged from 9440 to 9880 Btu per kwhr, and the amount of power consumed for station service ran between 4.7 and 4.8 per cent as compared to 5.5 or 6.0 per cent for comparable non-reheat machines on the same system.

During the month of June tests were made to determine the ability of these reheat units to trip out when carrying full load, enabling a study to be made of changes in primary and reheat steam temperatures and pressures, variation in reheater metal temperature, and action of control, intercept and safety valves. Load was first dropped from 25,000 kw, then 60,000 kw, and finally three times from 100,000 kw. In each case the load was dumped by hand-tripping of the main generator breaker, leaving the auxiliary load of approximately 2000 kw on the unit. The pulverizers are arranged to trip automatically when the turbine control valves go closed and remain closed for a period of three seconds. Automatic combustion control was in service at the time of tripout except during the first test. Immediately upon tripping of the pulverizers, all major equipment was moved to hand control.

On the basis of the tests, both the boiler and turbine controls are considered adequate in the event of an accidental loss of any or all load. The maximum speed reached by the turbine was 3840 rpm. Inlet metal temperature of the reheater increased approximately 30 deg F during a two-minute period for the 100,000-kw test and then fell off at a rate of 7 to 8 deg F per minute. Although water in the gage glass fell out of sight, it was restored two minutes after tripout. In all probability the time required to determine the cause of an accidental tripping of the unit will determine how quickly it can be restored to service. The tests substantiated the concept that for least disturbance to either boiler or turbine it is desirable to restore load as quickly as possible.

With respect to starting procedure it was found that there is no limitation on boiler starting pressure from the standpoint of reheat temperature. Pressure can be brought to 1500 psig for safety-valve testing with no reheat flow without exceeding 800 F on the reheater thermocouples. The total elapsed time from lighting off until full load is seven hours.

Discussion

Two cases were reported in which turbine stop and control valves were hopelessly frozen in non-reheat plants. A question was raised as to the advisability of having two valves of the same design in series in the reheat line. Weekly closing of intercept valves was recommended by one manufacturer, along with plans to design future units with two of these valves in parallel on reheat lines.

Concerning start-up of reheat units from the cold condition, one engineer commented that these occurred so infrequently that operating schedules can accommodate a fairly long starting time. On the other hand, the ability to make quick starts from the hot-bank condition is of considerable importance because loads may drop off late at night or on week ends, following which it is desirable to return the unit to the line as quickly as possible.

Experiences of all four utility companies provided evidence that reheat does not impair reliability and that manufacturers of boilers and turbines have succeeded in simplifying reheat controls. Hope was expressed that this trend toward reliability and simplicity could be continued.

[1] For a description of the station see "Features of Dunkirk Steam Station" by R. P. Moore which appeared in COMBUSTION, October 1950, pp. 40–48.

Reheat Boiler Progress

"A Progress Report of Reheat Boiler Operation and Design" was the topic of a paper by **W. J. Vogel** and **E. M. Powell** of Combustion Engineering-Superheater, Inc.[1] By the end of this year 13 of the 87 reheat units designed by the authors' company in the post-war period will have been placed in operation.

The reheat temperature of the units described is controlled by vertical adjustment of burners arranged for tangential firing. Since the primary superheater and the reheater have different temperature characteristics, it is necessary to provide supplementary control which has usually taken the form of a spray-type desuperheater to limit the temperature of the high-pressure steam at reduced loads. This control system was chosen to avoid the necessity of adding spray water to the reheat steam. Water added at that point would reduce the steam flow through the high-pressure turbine and sacrifice some cycle efficiency.

To maintain a constant temperature at the turbine throttle the heat added per pound of steam flow through the superheater must be held constant. As the load on the turbine varies, the feedwater temperature will also vary, thereby changing the ratio of heat input to the unit and gas flow over the superheater per pound of steam generated. The problem in the reheater is quite different in that the temperature and pressure of the steam leaving the high-pressure turbine decrease with a reduction of load which increases the heat per pound of steam to be added in the reheater. It should be apparent that no single furnace temperature characteristic can satisfy the needs of both the primary superheater and the reheater. In order to minimize the quantity of desuperheating water and to obtain the maximum cycle efficiency over a wide load range, the heating surfaces are proportioned to require the same furnace temperature at full load.

The authors described two series of tests conducted at the Dunkirk Steam Station of Niagara Mohawk Power Company. The first set was run primarily to determine the performance characteristics of the turbine over a wide range of ratings but also provided an opportunity to make numerous gas-temperature traverses and to study the performance of individual sections of the superheater and reheater. During the second phase of the tests furnace temperatures were measured while burners were positioned at various angles of tilt.

Other tests were run at Dunkirk to determine the effect of sudden loss of electrical load. The load prior to trip out was 100,000 kw; it was dropped to 2000 kw and following a short predetermined time delay restored to the original load. Maximum steam temperature change during the test was 60 deg F, while during the period when no steam was flowing through the reheater the temperature of these elements increased 40 or 50 deg F to a maximum of 790 F. This was well within the limits of safe operation.

The authors concluded with the statement that the combined operating experience, including numerous start-ups, trip outs and a variety of operating conditions, has justified the confidence that engineers placed in the ability of manufacturers to design reheat units which are at least as reliable as the standard straight-through unit.

In a paper entitled "Some Design Factors Relating to Performance and Operation of Reheat Boilers," **H. H. Hemenway** of Foster Wheeler Corporation discussed several means for controlling primary and reheat steam temperatures and cited some of the advantages and disadvantages of indirect reheating.

Since central stations must be designed for efficient operation at loads below full load, it is essential that full steam temperature be provided at partial load. A convection superheater has a temperature characteristic which rises with load, and with convection reheater and superheater surface the heat absorbed at full load will be greater than required. To absorb this additional heat requires more heating surface than necessary to produce control temperature. Increasing surface alone may not be sufficient, and it may become necessary to increase exit gas temperature with its potential slagging problems.

The excess heat to the superheater over the control range may be regulated by the use of heat-exchange devices for removing heat from steam, by spraying water into the steam, by causing some of the gas to bypass heating surface, or by differential firing.

One possibility of avoiding increase in exit gas temperatures is the use of the radiant superheater which absorbs part of the furnace radiation for raising steam temperature. At normal superheater velocities and flow rates the cooling effect in a radiant superheater is less than obtained in a waterwall tube, so that superheater metal temperatures must be kept within safe limits. That this design problem can be met is attested by the operating experience at the Port Washington Station of the Wisconsin Electric Power Co.

An exception to the condition that an increase in control range increases superheater duty at full load is the combination radiation-convection superheater. With this type the reheater may be designed with a wide control range while the furnace exit temperature is held at a relatively low level. This is possible because the heat removed from the gas in the convection superheater of a combination is only a portion of the total heat required for superheating.

Indirect reheaters to heat secondary fluids which in turn reheat steam between turbine stages have several advantages: (1) substitution of relatively short piping between turbine and secondary reheater for the large and expensive piping between turbine and boiler, (2) reduction in reheat pressure drop, (3) possibility of providing more reheat stages because of short connections, (4) adaptability to low pressure design for secondary fluid system, and (5) wide control of reheat temperature by varying flow rate of secondary fluid. Disadvantages include: (1) development of means of handling secondary fluid (molten salt or liquid metal) not yet at the commercial stage, (2) two sets of heating surfaces to replace one and (3) increased surface required because of reduced mean temperature difference.

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[1] This paper will appear in more complete form in a future issue of COMBUSTION.

P. R. Loughin and **H. H. Poor** of The Babcock & Wilcox Company presented a paper entitled "Operation and Performance of Modern Reheat Boilers" in which they discussed operating procedures for normal start-up, quick start-up, normal operation, and shutdown under normal or emergency conditions. In starting up from cold there is little difference between a reheat and a non-reheat boiler. The reheater should be drained as completely as its design will permit before lighting off. If it is non-drainable any water which may remain in the tubes must be boiled out during the starting-up firing period. Firing rate must be controlled during the pressure-raising period so that the gas temperature entering the reheater will not exceed 900 F before steam flow through the reheater begins. Unusually adequate protection for the reheater is provided if the normal precautions to protect the superheater are followed.

During normal operation the reheater will be protected against overheating provided that its outlet steam temperature does not exceed design values. At low loads it may be found economical to operate a reheat boiler with high excess air in order to elevate primary and reheat steam temperatures. Although boiler efficiency drops off, the overall cycle efficiency is improved because of the increased turbine efficiency with the higher steam temperatures.

With regard to temperature regulation, reheat steam temperature is even more sensitive to operating changes than is superheater steam temperature, since superheater and reheater heat absorption are both affected in the same way. In addition, superheater outlet temperature changes are passed along with only partial diminution as the steam passes through the high-pressure turbine to the reheater inlet.

Shutdown procedures for reheat and non-reheat boilers are the same except for the additional set of drains which should be opened after the turbine is taken out of service. Rate of cooling is limited by consideration of thermal stresses in the drum. In cases where it is desired to enter the boiler setting as soon as possible after the unit is taken off the line, the permissible rate of pressure reduction may be increased by raising the water level to the top of the drum and holding it there so that both the top and bottom will be cooled at the same rate.

If the unit is to be shut down temporarily in a hot-bank condition, the available means of steam temperature control may be used so as to minimize temperature shock to the turbines when they are returned to service. While the boiler is off the line it may be fired periodically at a rate such that gas temperature entering the superheater or reheater does not exceed 900 F. In this way boiler pressure may be maintained and accumulation of condensate in the superheater and reheater will be prevented. The reheater drains should be open while the boiler is on hot bank.

Reheat Turbine Progress

"Modern Reheat Turbines—Service Experience and Recent Design Progress" was the subject of a paper by C. **Schabtach** and **R. Sheppard** of the General Electric Company. They presented a report of the first year or two of operating experience with reheat machines originally described at the 1948 ASME Annual Meeting and mentioned some of the design features of units now under construction.

In July 1951 there were, among the units built by the authors' company, five 125,000-kw cross-compound reheat turbine-generators in operation in stations of the American Gas & Electric system, and ten tandem-compound, double-flow 3600-rpm reheat units in ratings from 50,000 to 80,000 kw running in stations of other utilities. Heat rate tests have been conducted on one of the A. G. & E. cross-compound units and on two tandem-compound machines at the Dunkirk Station of Niagara Mohawk Power Corp. Guarantees in all cases have been met or bettered. At Dunkirk loads of 100,000 kw were dumped three times. The speed of the unit reached a maximum of 106.5 per cent, dropping to 105 per cent within ten seconds. For about two minutes a 2000-kw auxiliary load was carried on the steam stored in the reheater. The unit was resynchronized with the system in about $4^1/_2$ minutes. The coal mills were automatically tripped through switch contacts in the high-pressure control-valve operating cylinder, and temperature of the steam from the reheater started dropping immediately after load dumping. Similar tests have been made on other units, and some accidental load dumps have occurred. In each instance the governing mechanism has performed satisfactorily.

Cold starts of reheat turbines with single boilers are made with steam at reduced temperature and pressure, permitting the turbine piping and the boiler to be brought up to temperature gradually and uniformly. Hot starts following shutdowns of short duration should be made with steam whose temperature is close to the temperature of the metal parts near the inlets. As far as the turbine is concerned, a reheat unit may be brought to speed as quickly as a non-reheat machine of comparable size and design. If the reheat and initial temperatures can be controlled with equal accuracy, the reheat turbine may also be loaded at the same rate as the non-reheat turbine.

The following design features are possessed in common by a number of reheat machines now on order for higher capacities, pressure and temperatures:

(1) Where both initial and reheated steam are admitted to the same casing, the admissions are immediately adjacent, with flow in opposite directions through the turbine to the reheater and condenser, respectively. The pressure difference between the initial and reheat inlets has not been the cause of a problem.

(2) Where the high-pressure section above the reheat point of tandem-compound turbine is in a separate casing, the direction of steam flow through this section is opposite to that through the section following the reheat point. The thrust bearing is located between the high-pressure and reheat turbine casings, which reduces differential expansion between rotating and stationary parts.

(3) Ferritic materials are employed for shells, pipes and valve bodies at 1050 F instead of austenitic materials used in the first reheat units built for this temperature. The latter are employed only for valves, valve stems and bushings, where their superior corrosion resistance is a necessity.

(4) Double-shell construction is used for the high-

pressure, high-temperature sections (both above and below the reheat point) when required by stress and temperature considerations.

(5) The control system is basically the same as that initially developed for tandem- and cross-compound reheat units.

The authors described design features of a tandem-compound unit with triple-flow exhaust for the Public Service Company of Northern Illinois and for the Kearny Station of the Public Service Electric and Gas Company, and cross-compound units for the Detroit Edison Company and the Astoria Station of the Consolidated Edison Company of New York.

In a paper entitled "Present Development of the Reheat Steam Turbine," **R. L. Reynolds** of Westinghouse Electric Corporation reviewed the advantages and disadvantages of the reheat cycle, cited its present degree of acceptability, and discussed a number of control and design features of reheat turbines. While no reheat turbines are now being built for capabilities of 50,000 kw or below, they may be justified for these smaller ratings if the trend toward higher load factors and higher fuel costs continues. The most common inlet steam conditions have been 1450 psig, 1000/1000 F, but an increasing number of machines are being purchased for 1800 psig, 1000/1000 F, and one is now on order for an initial temperature of 1100 F at 2350 psig and a reheat temperature of 1050 F. Careful consideration should be given to the economics of reheat for units of 60,000 kw and larger, load factors above 50 per cent, and fuel costs of 20 cents or more per million Btu.

The accompanying diagram shows how a reheat turbine may be controlled. The only significant differences between this system and that for a non-reheat turbine are the additions of a bypass valve (*5*), interceptor valves (*4*) with their control (*G*), and dump valve (*6*) with its control. The bypass valve is intended to avoid possible overheating of the low-pressure section during starting and to bypass the reheater when washing the turbine. However, experience with single boiler-turbine units has shown it unnecessary to open the bypass when starting. Interceptor valves are installed to shut off the flow of steam to the intermediate pressure element in case of sudden loss of electrical load. Two of these valves in parallel are furnished so that each may be tested separately while the unit remains in service carrying reduced load. A dump valve is provided on certain arrangements of reheat turbines to dissipate the energy of the steam passing through the diaphragm seals when the interceptor valves have tripped closed. When the dump valve is opened, steam flows directly from the intermediate pressure inlet to the condenser, bypassing the intermediate and low-pressure elements.

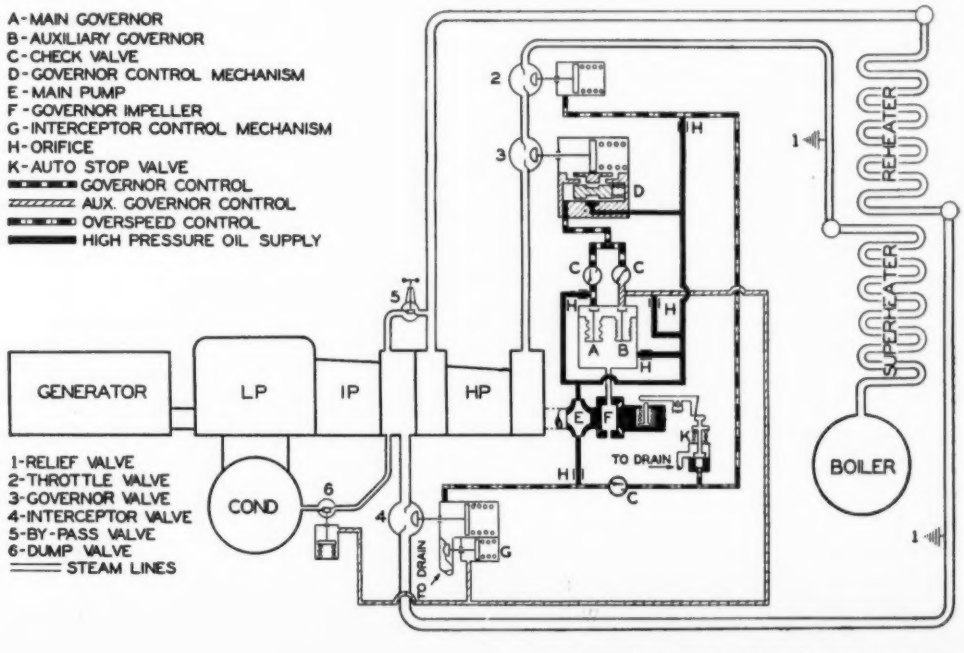
Discussion

The use of a second stage of reheating to gain a further 2 to $2^1/_2$ per cent in efficiency was mentioned as a possible future development.

Much of the discussion centered on the effect of iron oxide deposits in preventing complete closure of intercept valves. It was pointed out that reliable operation of the intercept valves is of the same importance to reheat turbines as control of main throttle valves. In starting up reheat turbines an important factor in reducing exhaust temperature is to have a good vacuum, and to accomplish this the installation of large priming ejectors was recommended. Cooling sprays are installed in the exhaust hood by some turbine manufacturers as added protection.

Steam Turbine Performance

In a paper entitled "Thermal Performance of Modern Turbines" **H. R. Reese** and **J. R. Carlson** of Westinghouse Electric Corporation provided data on heat rates that can be expected from turbine-generator units operating on the regenerative cycle with inlet steam conditions ranging from 850 psig, 900 F to 2400 psig,



Reheat turbine control diagram

1100 F. Similar information was provided for turbine-generator units operating on the reheat cycle.

Following an enumeration of standardized steam conditions the authors presented superheat correction factors, straight condensing heat rates, and turbine-generator heat rates applicable to various numbers of feedwater heating stages. Methods of calculating throttle flow, condenser flow and heater flow were also explained, as were corrections for initial pressure and temperature, number of stages of feed heating, exhaust pressure, and terminal difference in top heater. Calculations specifically applicable to the reheat cycle were shown.

The paper and its accompanying charts provides useful information for preliminary power plant design, extending the data originally presented in a 1945 Westinghouse bulletin entitled "Steam Power Plant Planning Guide."

C. W. Elston and **P. H. Knowlton** of the General Electric Company, in a paper entitled "Comparative Efficiencies of Central Station Reheat and Non-Reheat Steam Turbine-Generator Units," reported on their company's experience in testing large turbines under what are essentially service operating conditions. Results from such testing have enabled evaluation of the merits of design improvements, confirming progress in the art, and have provided a sound foundation of knowledge upon which the performance of new designs may be predicted. The present paper represents an attempt to bring up to date data presented in 1941 by G. B. Warren and P. H. Knowlton on "Relative Engine Efficiencies Realizable from Large Modern Steam Turbine-Generator Units."

Over the past six years machines built by the authors' company have followed these trends:

1. The average rating has doubled to roughly 60,000 kw in 1951.
2. The average pressure has increased from approximately 850 psig to 1150 psig with about 20 per cent of capacity installed this year in excess of 1450 psig.
3. The average temperature has increased from 875 to 950 F and the maximum temperature, from 950 to 1050 F.
4. Maximum capability of 3600-rpm turbines has increased from 60,000 to 125,000 kw with a unit of 200,000 kw scheduled for shipment in 1954.
5. This year 30 per cent of the turbine capacity being shipped utilizes the reheat cycle, and this percentage will be doubled in 1952.

An interesting chart provides a comparison of test results with guarantees for 3600 and 1800-rpm turbines tested in the past ten years. The spread between guaranteed and tested performance is made up of these factors:

1. A margin provided by the designer in guarantees to take care of the usual small residual uncertainties of manufacture and testing.
2. The incorporation of details from which performance improvements were to be expected but which were not fully capitalized when making performance guarantees.
3. The unusual as well as the usual inaccuracies in actual testing.
4. The uncertainties as to the actual internal condition of the turbine with respect to cleanliness and deposits.

Discussion

It was pointed out that the two papers complemented one another and dovetailed extremely well, considering two different and independent approaches. For a rapid estimate suitable for preliminary central station design the Reese and Carlson paper gives enough data to limit alternatives to three or four. The Elston and Knowlton paper tells more of internal design of turbines and factors that may be applied to determine efficiencies.

The desirability of having some yardstick to measure turbine cleanliness was mentioned. There is a need to know performance under clean conditions so as to evaluate "dirtying up" and to establish a point at which cleaning should be undertaken. It is understood that research is now under way with a dummy turbine nozzle through which the turbine steam is passed and upon which build-up of deposits may be observed. It was pointed out, however, that knowledge of the effect of deposits is incomplete and that turbines of the same design may perform with inexplainable differences.

Furnace Performance Factors

A. R. Mumford of Combustion Engineering-Superheater, Inc., and **R. C. Corey** of the U. S. Bureau of Mines reported on "An Investigation of the Variation in Heat Absorption in a Natural Gas-Fired, Water-Cooled Steam-Boiler Furnace." The investigation was carried out on boiler No. 9 at the Sterlington Steam Electric Station of the Louisiana Power & Light Company, using methods established by the ASME Special Research Committee on Furnace Performance Factors. Four series of tests were made, comprising 14 complete runs made at $^1/_4$, $^1/_2$, $^3/_4$ and full load and at low, medium and high excess air. Superheater outlet conditions for the boiler are 900 psig, 915 F, and normal continuous operation is about 380,000 lb of steam per hour. There are two rows of four ring-type gas burners having inside adjustable vanes and outside fixed vanes. For the tests 71 tube-surface thermocouples were installed, 20 on the front wall and roof, 18 on each side wall and 15 on the rear wall.

The absence of solid-particles of ash or slow-burning heavy hydrocarbons makes a natural gas-fired furnace free from effects which, with other fuels, mask the influence of the flame path and the relative area of the flame envelope. The tube surface temperatures on the furnace walls were essentially constant at any given rating and excess air and were completely reproducible. A check test made after an interval of more than a week gave surface temperatures within a degree or two of their previous values. One conclusion is that the shape of the invisible flame envelope was independent of load, at least with fixed burner position and fixed mixing angles. On previous tests this factor had not been apparent because of the masking effect of variable ash deposits on furnace tube surfaces.

In normal service the test boiler was used at high capacity and high load factor, and the mixing vanes and

flows in the burners had been properly adjusted for this condition. When the tests were conducted vane adjustment was not attempted for partial-load runs. In plotting isotherms it was apparent that the pattern was similar at all loads with four or eight burners in operation. This similarity and constancy of pattern indicated that, as long as burners and mixing devices were not readjusted, the hot gases would impinge on the walls at the same spots at all ratings. That is to say, the path of gases through a furnace is determined by the burner position and the geometry of the furnace and not significantly by rating or excess air, another factor not previously observed in tests of pulverized-coal firing.

With increase in excess air it was found that the temperatures of the gases leaving the furnace at constant steam load decreased although the heat content of the mixture increased. The patterns of the isotherms of gas temperatures at the furnace outlet varied markedly with the load.

Cyclone Furnaces

Two companion papers were presented on cyclone firing, the first on "Station Design with Cyclone-Fired Steam Generators," by **H. C. Schroeder** and **R. J. Strasser** of Sargent & Lundy, consulting engineers; and the second on "Operating Experiences with Cyclone-Fired Steam Generators," by **V. L. Stone,** superintendent of generating stations of the Commonwealth Edison Company, and **I. L. Wade** of the Public Service Company of Northern Illinois.

The first of these papers, after reviewing details of several installations, concluded: that the ash handling is no more complicated than with pulverized coal; coal preparation cost is less and smaller dust collectors are required; also that less space is required. However, despite the lower dust content of the gas, the dust with Central Illinois coal is of such composition that it sticks to the heat-absorbing surfaces with the result that to date there has been no reduction in equipment and labor for cleaning the surfaces when burning this coal. The authors expressed the hope that use of tempering air and new methods of cleaning may be helpful in this respect.

The second paper described and reviewed some five years' experience with cyclone firing in five plants of the Commonwealth Edison Company and the Public Service Company of Northern Illinois, which normally burn Central Illinois coal. This runs high in ash and has a low ash-fusion temperature. With this fuel it was found that the cyclone slag tap is sometimes fouled with coke and viscous slag during the bring-up, but after high rating has been established the tap clears itself. The practical load range is approximately 50 to 110 per cent of rated capacity.

Natural gas and a variety of fuel oils have been burned successfully in several of these furnaces, alone and in combination with coal. Also, a number of coals, ranging in rank from North Dakota lignite through Pocahontas and coals from France, have been tested. With some it was necessary to deviate from normal operation, such as employment of very high air preheat, but only in the case of Pennsylvania Top Freeport coal was the performance inacceptable owing to its high ash-fusing temperature and viscous character of slag.

At Fisk Station, which is direct-fired, maintenance costs, including the Redler conveyors, feeders, crushers, burners, etc., are about seven cents per ton. Cyclone wear occurs mainly in the inlet scroll and on the cone surface. Life of alloy cast-iron or alloy cast-steel scroll liners is about 30,000 tons for the more severe regions, although six carbide-faced liners, now in service over a year, have so far handled some 55,000 tons of coal. While there have been some cyclone tube failures, none were attributable to circulation or boiler-water conditions.

Experience has shown a saving in power required to prepare the coal for firing, but that this saving is more than offset by an increase in forced-draft fan power.

With reference to dust emission, the carbon content of the fly ash was given as 1.5 to 10 per cent under normal conditions and as high as 15 to 20 per cent under poor firing conditions. Tests at Fisk have shown 82 to 90 per cent of the ash in the coal to be removed as molten slag.

The paper devoted considerable space to superheater fouling difficulties encountered at both Fisk and Joliet stations and described the steps taken to combat them.

Discussion

The discussion centered largely around slagging of the superheater near the furnace outlet and the power requirements.

Since soot blowers proved ineffective in removing the slag, experiments were tried of shooting marbles at high velocity into the slag clogged between the superheater tubes, but this was found to mark the tubes perceptibly.

Also, gas recirculation is being tried. The gas is taken from the economizer outlet and introduced close to the furnace outlet in order to reduce the temperature of the gases entering the superheater.

In reply to a question as to power consumption, the authors stated that the fan power amounts to about 44 kwhr per ton of coal. Against this is the saving in power required for coal preparation through elimination of the pulverizers. Hence the net excess power of the cyclone unit, as compared with a conventional pulverized coal unit in the same station, amounts to 15 or 17 kwhr per ton.

Experience has indicated the most satisfactory operating pressure for the cyclone to be 30 to 40 in. of water. Below 30 in. combustion is not as good as above 40 in.

Expansion of American Gas and Electric System

A session was given over to three companion papers dealing with "The Development and Implementation of a Generation Program of the American Gas and Electric Company." The first, under the authorship of **Philip Sporn,** dealt with system fundamentals; the second, by **Mr. Sporn** and **H. A. Kammer,** discussed fuel supply; and the third, by **Mr. Sporn** and **S. N. Fiala,** described the two latest plants of the system—the Kanawha River and the Muskingum River Stations, now under construction, respectively, for the Appalachian Electric Power Company and the Ohio Power Company.

This system serves 2165 communities having a total population of 4¼ million in highly industrialized areas of portions of Michigan, Indiana, Ohio, West Virginia, Virginia, Kentucky and Tennessee. The energy input is now at the annual rate of 16 billion kilowatt-hours

and is expected to reach 20.2 billion by 1954, with a projected demand of 3,400,000 kw. In addition to a number of older plants which have been extended from time to time, the major post-war developments include five large stations, namely, Tidd, Philip Sporn, Tanners Creek and the new Kanawha and Muskingum developments. All are coal-burning plants.

The rapid industrial growth of the area served has made necessary new concepts of large-scale energy generation with regard to size of units, as well as close attention to fuel costs. With respect to the latter, Mr. Sporn pointed out that a difference in production costs of as little as one-tenth of a mill per kilowatt-hour involved differences in annual operating costs of around two million dollars a year with a system input of 20 billion kilowatt-hours. Hence, adequacy of coal supply, with minimum transportation and handling expense, has been an important factor in site selection. At present the major plants of the system are consuming more than 8 millon tons of coal per year, which figure will increase to 9,140,000 tons by 1954, when approximately half of the 3,600,000 kw installed capacity will be operating on strip-mined coal.

The new Kanawha plant, located on the Kanawha River about 22 miles upstream from Charleston, West Virginia, and the Muskingum plant on the Muskingum River some 46 miles from Zanesville, Ohio, will be similar in most respects. The principal difference will be that the boilers of the former will have dry-bottom furnaces and those of the latter wet-bottom furnaces, because of the low ash-fusion coal to be burned.

The initial installation for each plant calls for two 200,000-kw units and an ultimate installation of five. This capacity rating refers to the net effective salable capability at the high-tension bus bars after deduction of all losses and station uses. The first unit is scheduled for service early in 1953.

Each unit will comprise a cross-compound turbine-generator supplied with steam at 2000 psig, 1050 F by its own reheat boiler which will reheat the steam for return to the high-pressure machine at 460 psig, 1050 F.

The decision to employ units of 200,000 kw was largely dictated by the fact that, while the present system peak is now in excess of 2,480,000 kw, by the time the fourth of these initial machines is on the line the peak will have increased to around 3,200,000 kw. This rise is coupled with a system load factor of 70 per cent. Experience of the company has shown that whenever the ratio of the net capability of any extension to that of the system falls to 5 per cent it is desirable to increase the size of the next units to be installed. Moreover, as Mr. Sporn pointed out, the use of such large units makes possible savings in operating labor and supplies, as well as in initial investment for boilers, turbines, piping, building and substructures.

With all equipment housed, except the fans, air heaters, dust collectors and connecting ductwork, the building volume figures 19.3 cu ft per kw, including machine shop and offices, to house two 200,000-kw units. This compares with 26.4 cu ft for the first two 150,000-kw units at the Sporn Station. At a cost of 75 cents per cubic foot, this means a saving of over a million dollars per 200,000-kw unit.

Each boiler is designed for a furnace heat input of 1.84 billion Btu per hour, with a primary steam flow of 1,335,000 lb per hr and a reheater flow of 1,216,000 lb per hr. It will be provided with eight mills of which seven can carry full load and consume approximately 80 tons of coal per hour. Three Ljungstrom air preheaters will heat the combustion air to 550 F. The boiler casings are being designed for pressurized furnace operation, although there will be three induced-draft fans, as well as two forced-draft fans per unit.

Superheat and reheat temperature control below full load will be obtained primarily by recirculation of flue gas from the economizer outlet to the furnace, with water spray attemperation at the reheater inlet and between stages of the superheater for critical control.

These new stations are expected to show a heat rate of around 9000 Btu per net kilowatt-hour which, with fuel costing 16.5 cents per million Btu, should result in a unit fuel cost of less than 1.5 mills per kilowatt-hour.

Discussion

E. H. Krieg complimented Mr. Sporn on looking far ahead as to coal supply as a most important factor in selection of plant site.

Elmer Kaiser asserted that the coal industry is ready to meet all requirements for many years and that the only bottleneck in some instances has been transportation. The use of new type machines in mechanized underground mining promises to speed up both production and conveying to an extent that may have considerable bearing on future cost.

W. H. Rowand mentioned that boiler studies indicate single steam generating units of greater than 200,000 kw to be entirely feasible, to which **Glenn Warren** added that turbine builders are prepared to go beyond 200,000 kw when utilities are ready.

Another discusser observed that with growing government limitations on profits why should utility engineers continue to seek greater efficiency; also that it was difficult to reconcile the low fuel costs mentioned in Mr. Sporn's paper with the refinements adopted to secure such a remarkably low projected station heat rate of slightly over 9000 Btu per kwhr. To this question Mr. Sporn replied that there were factors other than fuel cost entering into the design problem; furthermore, that low price fuel may not continue, in which case efficiency would predominate.

Panel on Light-Load Operation of Spreader Stokers

Max O. Funk of Combustion Engineering-Superheater, Inc., stated that the problem of light-load operation of spreader stokers resolves into one of maintaining reasonable minimum grate heat releases for adequate temperatures and excess air at the grate line. In selection of equipment, maximum load conditions should be chosen so that an excessive range will not penalize low-load operation. Oversized equipment reflects itself in higher minimum conditions. For a given range of steam demand it may be necessary to use more units of a smaller size to achieve maximum requirements while at the same time staying in an acceptable range for minimum requirements.

Design factors affecting low-load operation include continuity of coal feed (to eliminate a cycling grate re-

lease), control of air and coal distribution so that grate areas may be reduced, and overfire jet arrangement. Mr. Funk reported on a spreader-stoker-fired installation successfully operating through a 9 to 1 range and maintaining a stack discharge within the limits of a local smoke ordinance similar to the ASME Model Smoke Law.

Herbert L. Wagner of Detroit Stoker Company took issue with the contention that fly ash and carbon loss in the gas stream are proportional to some power of the burning rate. Instead, he pointed out there are sufficient test data available to show that such carryover follows a flat curve in relation to the burning rate. He added that, regardless of burning rate, it is impossible to meet present-day ordinances without some types of dust-arresting equipment. Therefore, there is no sense in making grate areas very large in an effort to keep the burning rate exceptionally low at maximum continuous load, as such units are likely to simulate smudge pots at light loads.

Features of the present-day spreader stoker that can be applied to produce smokeless operation during light-load periods were listed by Mr. Wagner. Briefly, these were as follows:

The maximum burning rate should be in keeping with the type of grate. For the dumping type a heat release of 350,000 to 600,000 Btu per sq ft per hr, depending upon the ash in the fuel, is practical; and with high-ash coals it is often necessary to reduce the time between cleaning periods. For the continuous-discharge type heat releases between 700,000 and 800,000 Btu per sq ft per hr may be employed and little attention paid to the ash content of the coal.

The number of feeders on the dump-grate type is most important and is influenced by the percentage of ash in the fuel—the greater the number of feeders the better the cleaning conditions and the better the fuel distribution crosswise of the grate.

The pattern of overfire air to create furnace turbulence is most important. In some cases it is desirable to supplement the high-pressure overfire air with steam jets for use under extremely light load conditions, at which time the furnace is loaded with excess air and temperature is low. Also, the elevation and lateral spacing of the cinder returns have important bearing. They should be injected into the high-temperature zone in such a way as to maintain uniform distribution.

Longitudinal zoning on the continuous-discharge stoker is one method of coping with light-load conditions. Also, lateral zoning of the rear portion is helpful where the expected turn-down ratio in load is in the bracket of from 6 to 1, to 12 to 1.

Finally, the size consist of the fuel plays an important part in smokeless operation, the ideal size being $^3/_4$ in. by 0 with one-third $^3/_4$ in. by $^1/_2$ in., one-third $^1/_2$ in. by $^1/_4$ in., and one-third $^1/_4$ in. by 0.

D. J. Mosshart of Westinghouse Electric Corporation considered the two major problems involved in the operation of spreaders at low combustion rates as: (1) maintaining operation with excess air at a value sufficiently low for good efficiency; and (2) operation without objectional smoke.

The problem of minimizing excess air is one of controlling air admitted over the fire for such items as cooling of feeders, dispersal of fine fuel at the feeder openings, return of cinders to the furnace, and secondary air jets. This also constitutes a major portion of a solution to the smoke problem as experience shows that an increase in excess air beyond certain limits results in an increase in smoke.

The basic cause of smoke at low combustion rates seems to be in the decomposition, or cracking, of the hydrocarbon vapors released from the coal before these vapors are mixed with air in quantity sufficient to support their combustion. If the temperature in the fire-bred region is relatively low, the decomposition appears to occur without complete combustion and particles of free soot will be seen in the furnace gases.

Mr. Mosshart recommended providing two sets of overfire air jets—a row of small nozzles in the rear wall about 12 in. above the grate, and a row of large nozzles in the same wall about 3 or 4 ft above the grate. The nozzles should be so sized that, at full air pressure, one-quarter of the total rear wall secondary air will flow through the lower nozzles and three-quarters through the upper. Flow through the lower nozzles should be fixed, but that through the upper nozzles should be controlled automatically; and shut off, except for a small flow for nozzle cooling, at less than 25 per cent of full boiler load.

Although steam jets, instead of air jets, to provide turbulence are an aid to smokeless operation with somewhat lower excess air, the author regarded their use as economically intolerable except where low-load operation is only occasional.

F. C. Messaros of American Engineering Company, mentioned that occasionally a buyer will specify higher peak capacities than are really required, which results in excessively low burning rates on the grate at minimum load. This has led, in some instances, to the stoker manufacturer gambling on the real load required and on providing too little grate area for the specified peak; in which case smoke at low load is minimized. But if the specified peak is correct, excessive burning rates may result.

Adequate furnace volume is most important, as is also overfire air to control smoke at higher rates of burning; but the latter becomes of less value with decreased burning rates. Very small units may operate satisfactorily with refractory walls and without direct overfire air, if the fuel distribution on the grate is good. In such cases the setting leakage provides sufficient overfire air with the high-temperature refractory walls assisting ignition, while the carrying air for cinder return provides adequate turbulence.

Uneven discharge of coal from the feeders with consequent uneven spreader of fuel on the grate tends to produce smoke.

Mr. Messaros believed that more attention should be given excess air at all ratings, as many spreader-stoker installations bristle with overfire air and cinder-return nozzles, all pumping air into the furnace. Much of this air may be of little value in producing turbulence, but quite detrimental to performance.

There has always been a question as to the merits of returning the fly ash to the furnace at low loads and

proper analysis of the problem for each case will reveal that often such return is not justified. In any event it would appear advisable to arrange the return equipment so that it may be cut out on extended low-load periods.

Recent trends indicate an increase in the number of installations using a decantation type of dust collector in which only the coarser particles, high in carbon content, are rejected and the finer particles returned.

E. C. Miller, of Riley Stoker Corporation, attributed low-load difficulties with spreader stokers to a number of factors including coal characteristics, distribution of coal and air, uniformity of coal size or segregation, combustion control action, grate burning rates, furnace turbulence, and furnace temperature. Correction of some of these is the responsibility of the designer whereas others are operational

Refractory furnaces, he said, give relief at low ratings but are a handicap at high ratings; whereas preheated air, use of furnace gas for turbulence and reinjection, and the rear-discharge traveling grate are effective tools for improving low-load performance; besides being beneficial for high loads.

Discussion

This symposium brought out a number of discussions from the floor, the high spots of some of which may be summarized as follows:

1. In some instances users are burning double-screened coal instead of nut and slack in an attempt to avoid smoke; also many small installations lack combustion control and cinder recovery equipment.
2. Low load operation at heat releases down to 100,000 Btu per sq ft per hr can be had without smoke; and, although zoning of the grate is most desirable, a 6 to 1 load range, without zoning, will be satisfactory.
3. Some of the low-load troubles could be minimized through employment of arches, especially the long arch as frequently employed with regular traveling grate stokers. In one case cited good results were attained with a bridgewall when burning a very poor grade of fuel.
4. Maximum turbulence at the grate level is most important in order to minimize smoke at low loads.
5. While one speaker mentioned an investigation being conducted on a large stoker involving a load range from 160,000 to 1,000,000 Btu per sq ft per hr with overfire air, another suggested that for very low loads it might in some cases be desirable to switch to oil.

Gravity Reinjection of Fly Ash

A paper under the authorship of **C. H. Morrow, W. C. Holton** and **H. L. Wagner** described an investigation of gravity reinjection of fly ash at the rear of the furnace from the elevated dust-collector hoppers at the Racine plant of J. I. Case Co.

The unit was a four-drum boiler of 100,000 lb per hr rating having an economizer and tubular air preheater, and fired by a traveling grate spreader stoker, with travel toward the front. The grate is divided at four lateral sections and divided at the center line of the furnace and stoker. There are five rear-wall overfire air jets set $5^1/_2$ ft above the grate and four set 15 in. above the grate; also, six front-wall jets are set 5 ft above the grate and six more 10 in. above the grate. Following usual practice, fly ash and cinders collected in the hoppers under the economizer and under the stoker are returned to the furnace by pneumatic means.

Results of the tests showed a maximum dust loading at the stack of only 0.3 lb per thousand pounds of gas, compared with the model ASME Code specifications of 0.85 lb.

Gravity reinjection of the larger particles, such as collected in the hoppers under boiler and economizer, did not produce a significant reduction in dust loading from the furnace over that found when reinjecting by air.

Since it was not feasible to run a test with total fly ash reinjection by air, definite conclusions could not be drawn as to the relative merits of gravity vs. air reinjection of the predominantly fine dust particles from the dust collector. However, the large quantity of fly ash returned by gravity during the test did not cause any difficulties, such as slagging of furnace walls, clinkering of the stoker grate, or interference with the ability of the stoker to operate at 16 to 18 per cent excess air.

Correlation of Silica Carryover and Solubility Studies

Experimental studies as to the effects of different boiler-water constituents on silica carryover, in the range of 400 to 3000 psi, were reported in a paper by **C. Jacklin** and **S. R. Browar,** both of National Aluminate Corporation. These investigations were conducted on a laboratory boiler and indicated certain trends, as follows:

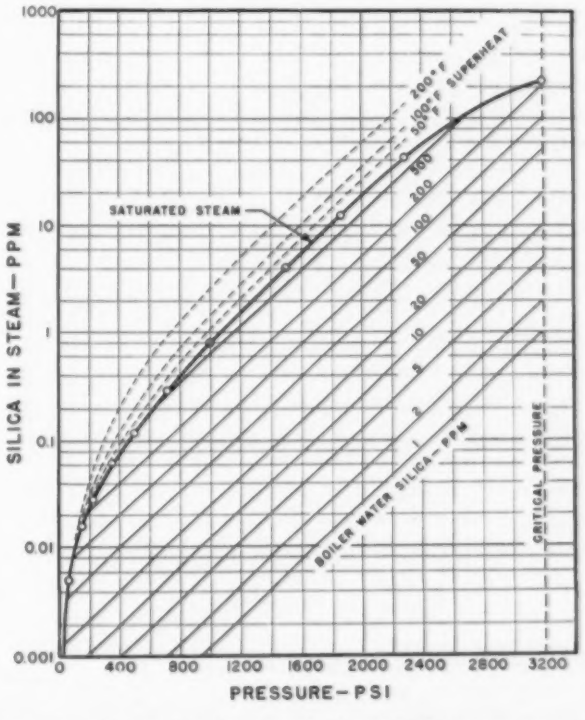


Chart showing approximate relationship between pressure, boiler-water silica, silica carryover, and silica solubility in saturated and superheated steam.

High ratios of hydrate to silica in the boiler water tend to reduce the ratio of silica in the steam to silica in the boiler water. The presence of calcium sludge, magnesium sludge, antifoam, organic treatments and various combinations of these materials in the boiler water caused no consistent increase or decrease in the ratio of silica in the steam to that in the boiler water. In fact, previous laboratory studies by other investigators were largely confirmed and the results correlated in a chart showing the approximate relationship between boiler pressure, boiler-water silica, silica carryover, and silica solubility in saturated and superheated steam.

Steam Contamination

"Although commercially acceptable steam purity can be obtained from boilers operating under widely differing conditions, there still remains the problem of removing the very small amount of impurities in the steam which cause troublesome turbine deposits. Further new developments and refinements in existing separating and purification equipment are necessary to eliminate this problem." These were the conclusions of a paper dealing with the influence of boiler design and operating conditions on steam contamination by Messrs. **P. M. Brister, F. G. Raynor** and **E. A. Pirsh,** all of Babcock & Wilcox Company.

The paper called attention to the fact that the ratio of silica to other contaminants is greater in steam than in boiler water and cited researches pointing to the vaporization of silica in high-pressure boilers as the explanation. Methods of separation and purification include (1) gravity or natural separation; (2) kinetic separation, including baffles or change in direction of flow and centrifugal force; (3) electrostatic precipitation; and (4) sonic agglomeration. The last two have not yet received practical application. At present, washing of the steam with relatively pure water appears to be the only practical way of removing silica from the steam.

Under the heading "Design of Separating and Purification Equipment," the paper described various baffle types, the cyclone separator and methods of steam washing. Drum internals, in addition to their steams separating functions, must allow access through the drum and have a minimum number of component parts.

Experience has shown that troublesome turbine deposits may occur with steam of less than 0.6 ppm total solids. In the 500- to 900-psi range, the turbine deposits that occur are usually soluble and can be removed by water washing; but in the 1000- to 1500-psi range they are the predominantly insoluble. However, operating experiences at 2000 psi and above have not thus far shown troublesome insoluble turbine deposits, although soluble deposits, readily removable by water washing, have occurred in this range in some cases. If the silica concentration in the boiler water does not exceed the values in the following tabulation, experience indicates that it will not be troublesome:

Operating pressure, psig	Max. silica conc., ppm
600–800	25
801–1000	15
1001–1500	5
Over 1500	3

Discussion

P. B. Place, of Combustion Engineering-Superheater, Inc., recalled that in earlier boilers having limited drum internals, carryover generally increased with increases in rating and concentration because the spray carried by the higher velocities was not separated; whereas modern internals remove practically all the spray. Thus the moisture content in the steam could conceivably remain constant over the operating range, but the impurity content in such moisture should change with boiler water concentration. Yet in rare cases boiler water concentrations of over 10,000 ppm have been maintained in boilers having relatively simple drum internals, whereas in other cases severe foaming has developed with concentration of less than 600 ppm.

However, the arbitrary concentration limits recommended by the ABMA represent average limits that can usually be attained with suitable drum internals.

Mr. Place observed further that with severe foaming, creating a blanket of foam on the water, a false water level may be indicated; but with control of foaming, either by low boiler-water concentrations or by suitable internals, water level can be varied quite freely without effect on steam purity, up to the point of flooding the internals at their steam outlet elevation.

Silica Determination

Two reports were made concerning research on silica determination conducted at the Armour Research Foundation of the Illinois Institute of Technology under the sponsorship of the Edison Electric Institute, The Association of Edison Illuminating Companies and leading boiler and turbine manufacturers. This is part of a cooperative research program designed to learn more of the mechanism of steam contamination and turbine-blade deposits.

The first report, entitled "The Spectrophotometric Determination of Small Amounts of Soluble Silica in Water," was prepared by **H. E. Robinson, E. A. Pirsh** and **E. Grimm.** The proposed method is sensitive to silica concentrations in steam condensates and boiler waters of 0.01 ppm, which is below the limit of 0.03 and 0.05 ppm at which silica concentrations are believed to cause turbine-blade deposits. The method recommends the use of the reduced silicomolybdate complex, the color being measured by a spectrophotometer at a wave length of 815 millimicrons. The use of 100-mm cells in the spectrometer, silica-free reagents and optimum color development permit sensitive analysis for silica in the 0.01-ppm concentration range.

Steam from modern high-pressure boilers will usually have a silica content between 0.01 and 0.20 ppm. Although some operators maintain boiler-water concentrations of 5 ppm or below, silica in boiler water will generally vary between 1 and 15 ppm. Various interfering ions and coloring matter may be present in boiler water, but condensed steam samples rarely contain interfering ions in objectionable concentrations. The investigation was mainly concerned with the determination of silica in condensed steam, although the procedures are also applicable to most boiler waters.

Because of the almost universal occurrence of phosphate ion in boiler waters it was necessary to study the interfering effect of this ion and methods for its elimination. Oxalic acid was chosen as the most effective means of eliminating phosphate interference.

The authors presented a detailed description of the method, including apparatus, reagents, and procedures for steam condensates and high-phosphate boiler waters.

In the second report, "Adaptation of the Spectrophotometric Determination of Small Amounts of Soluble Silica in Water to the Determination of Undissolved Forms of Silica," **H. E. Robison, E. Grimm** and **C. Brown** described an investigation of several methods for solubilizing unreactive silicates in boiler waters and steam condensates so that these forms may be detected by the spectrophotometric method previously described.

The first consideration in the investigation was the choice of an insoluble silica standard resembling the types of silicates most likely to be encountered in boiler waters or steam condensates. A number of sources of insoluble silica were investigated for stability of the dispersion and degree of insolubility. A specially prepared bentonite dispersion was found to be the most suitable for a source of insoluble silica.

Three methods of determining unreactive silicas have been developed: a potassium carbonate fusion in a gold crucible, pressure digest with caustic, and hydrofluoric acid solubilization. The fusion and pressure digest methods seem to be inherently less accurate than the hydrofluoric acid solubilization method but are useful as independent check methods.

Considerable emphasis must be placed on the cleaning and conditioning of the plastic vessels used in contact with the hydrofluoric acid. Cleaning procedures were found for the apparatus used that would result in low and consistent blanks. At present the method of hydrofluoric acid solubilization appears to be capable of analysis for total silica at 0.082 ppm with an accuracy of about four per cent.

An analysis for a form of silica of intermediate reactivity has been developed. It is done by means of a caustic digestion on a steam bath and has almost the accuracy of the soluble silicomolybdate analysis described in the first report.

A comparison was made of the soluble silica detected in various water samples before and after carbonate fusion and hydrofluoric acid solubilization. Little significant difference was found in the soluble and total silica in these waters, indicating that all the silica was soluble.

Steam at 2000 Psi, 1250 F

F. G. Ely and **F. Eberle,** of Babcock & Wilcox Co., reported on an investigation of the properties of new alloys for high-temperature superheater use as conducted on a 2-in. O.D. test element inserted in one of the Twin Branch boilers operating regularly at 2000 psi, 950 F. The element under test received steam at 950 to 1000 F from the superheater outlet and was so placed in the gas path as to raise the steam temperature to 1200 or 1250 F. To date some 5000 hr of operation have been accumulated of which one-third has been with a steam outlet temperature in excess of 1200 F, another third in the 1100 to 1200 F range and the remainder at temperatures below 1100 F.

The test element was in the form of four parallel hairpin loops connected in series, with legs of different materials. Eight alloys of the following composition were employed:

Alloy	Cr	Ni	Mo	Other Elements
Croloy 2¼	2.25	..	1	....
Type 304	18	8	...	....
Type 347	18	8	...	Cb
Type 321	18	8	...	Ti
Type 316	16	13	3	....
Type 318	16	13	3	Cb
Timken 16-25-6	16	25	6	....
Armco 17-14 Cu-Mo	17	14	2.5	3Cu-Cb-Ti

Field and laboratory inspection of the test element showed all the materials to be in generally good condition although subsequent periodic measurements of tube diameter indicated development of a small rate of creep at the hotter end of the 18-8 Ti (Type 321) material amounting to approximately ¾ per cent of tube diameter enlargement.

It is planned to continue this test program for an indefinite period. While it is too soon to draw definite conclusions, the authors point to the favorable indications for practical construction of commercial superheaters for 2000 psi, 1200 F.

Behavior of Superheater Materials at 1350 F

With steam temperatures of 1050 F in many new central stations and indications that 1200 F may not be too distant, the ASME Special Research Committee on High-Temperature Steam Generation undertook sponsorship of a program to evaluate commercially available alloys as to their suitability for superheater and reheater tubing at metal temperatures up to 1350 F, it being assumed that temperature of the heat-receiving surfaces will be from 50 to 150 deg higher than that of the steam.

The program was divided into two parts—one concerned with the reaction of high-pressure, high-temperature steam of such materials, and the other with corrosion by flue gases and slag-forming ash on the external surfaces. A paper dealing with the field test phase of this program, as carried out by Battelle Memorial Institute, was given under the authorship of **H. A. Blank, A. M. Hall** and **J. H. Jackson,** all of that organization.

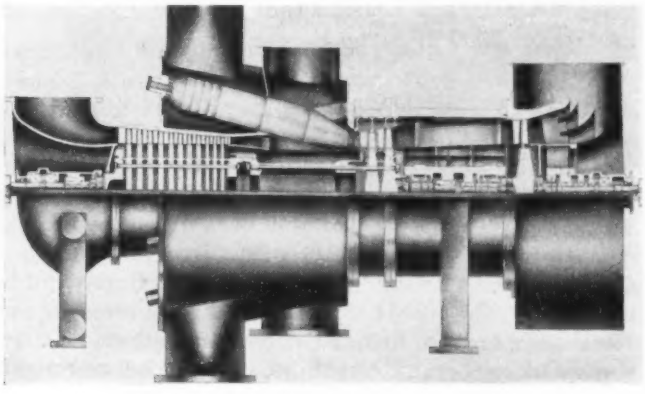
Fourteen commercially available alloys were selected and specimens inserted in racks that were located in steam generating units of ten companies burning various fuels. While the results varied considerably, due to differences in the atmospheres produced by combustion of the several fuels, they pointed to certain definite conclusions. These were that low-sulfur coal and gas produce combustion products that are only mildly corrosive, whereas high-sulfur, high-alkali and high-vanadium fuels produce combustion products that are often

extremely corrosive. In other words, the degree of corrosiveness appears to be related to the sulfur, vanadium and alkali metal content of the fuel.

Excluding considerations of strength and resistance to internal corrosion, it may be said that under mild corrosion conditions a variety of alloys is available for superheater tubing to operate at metal temperatures up to 1350 F without appreciable external attack. Under very corrosive conditions, all the alloys tested were corroded. Among them, however, AISI 309 S, 310 and 314 usually showed the better resistance to both general corrosion and subsurface attack.

Gas Turbines for Gas Pipeline Pumping

Savings in the form of reduced capital investment and lower transportation costs, through use of gas turbines for major gas transmission systems with lines 20 in. and longer, were reported by **T. J. Putz** of Westinghouse Electric Corporation. Additional advantages claimed were station simplicity, freedom from large water supply problems, reduced maintainence and ease of adaption for remote control or unattended operation. Also, he pointed out that the gas turbine's characteristic of increasing power with low ambients gives it extra rating in the winter months when the demand for gas is



Section through 5000-hp gas turbine

greatest, and the overall suitability of the characteristics of the gas turbine centrifugal compressor combination of high flow, low pressure ratio makes it ideal in booster service.

The present mileage of main gas transmission pipelines in the United States is approximately 100,000 miles with an installed compressor capacity of approximately three and one-half million horsepower. Currently there are over 17,000 miles of gas pipelines 25 in. diameter and above, either installed or authorized, and the total annual consumption of natural gas in the United States is now over five trillion cubic feet, not counting that used for repressurizing, field fuel and losses. Approximately 35 per cent of this gas is used in refineries and in producing carbon black; the other 65 per cent being pipeline gas. Various authorities have estimated that the amount of gas transported through pipelines will be doubled in the next 10 to 20 years. To handle this increase, a conservative estimate shows that an addition of 4,000,000 compressor horsepower will be needed.

"The simple, single-shaft gas turbine unit," said Mr. Putz, "is ideal as a booster in locations where great flexibility is not required and its use restricted to taking care of peak loads. Where more flexibility is needed to take care of large pipeline flow variations, the dual shaft gas turbine is required. With the addition of a regenerator, this unit becomes competitive on a fuel and lubricating oil cost basis with conventional direct-drive reciprocating gas engine units."

While a large number of gas-turbine-driven centrifugal compressor units have been reported sold, Mr. Putz said that, as of this fall, there was only one in actual operation in pipeline pumping. In May 1949 his company placed in service an 1800-hp gas turbine driving an Ingersoll-Rand direct-connected centrifugal gas compressor in a test station on a natural gas line at Wilmar, Ark. This was the first such installation in the world and the first industial gas turbine in the United States operating with natural gas as fuel.

Based on the operating experience with the 1800-hp single-shaft unit in pipeline pumping service and the results of studies of transmission line pumping requirements, the author's company has designed and is building a 5000-hp dual-shaft gas turbine with regenerator, a cross-section of which is shown. The unit will have an 11-stage axial flow compressor, six combustors, a two-stage compressor-drive turbine and a single-stage power turbine. This arrangement permits adjustment of power-turbine and gas-compressor speed and load to suit varying pipeline flow conditions. The regenerator which recovers approximately 75 per cent of the available heat in the exhaust gases is of the extended-surface type.

New 5000-Hp Gas Turbine

A second paper on gas turbines was that by **B. O. Buckland** and **D. C. Berkey,** of General Electric Company. This dealt with the design features of a new 5000-hp unit several of which are now being built, the first for delivery early in 1952. It is a one-compressor, two-turbine unit that can be used with or without a regenerator and can be readily applied to various arrangements involving the combined use of the steam and gas-turbine cycles.

The plant is designed for maximum flexibility and is intermediate in efficiency and complication between a simple one-compressor, one-turbine, non-regenerative power plant and a two-compressor, two-turbine, intercooled and regenerative plant. Since the load turbine is not required to drive the compressor element, the load speed can be chosen over a wide range of values without affecting the capability of the unit.

The axial flow compressor has 14 stages and operates at a pressure ratio of 5.5. The turbine has two stages, the second of which produces the useful load, and has a nozzle arranged so that its effective flow area can be varied during operation. The six combustion chambers are expected to operate at 800 F preheat and at 650 deg F rise.

It is expected that the plant will be particularly applicable to gas-line pumping or driving compressors for repressuring oil wells; also for industrial power generation of 3500 kw.

Petroleum Supplies

Adam K. Stricker, Jr., of General Motors Corporation presented a paper entitled "Impact of Defense Activities on Petroleum Supplies" in which he surveyed the principal supply and demand factors influencing the availability of crude supply and the overall refining capacity of the industry. He also examined some of the special problems affecting the principal petroleum products.

With respect to crude oil, American domestic production in June of this year was at an annual rate equivalent to the world production in 1941. The total Middle East production in 1950 was almost twice that of the Iron Curtain countries, but about one third of this has been temporarily lost with the closing of the Abadan refineries. A spread of the difficulty in Iran to other Middle East countries could have serious repercussions; already some of the crude from the Caribbean area, normally sent to the U. S. east coast, has been diverted to Europe.

More than 43,000 oil and gas wells were drilled in 1950, one third of which were dry holes. Off-shore drilling has advanced at a very rapid rate and is now taking place in several locations beyond sight of land, thereby adding to the nation's petroleum reserves.

The greatest untapped known oil reserves are in the Middle East. Iran, Saudi Arabia, Kuwait and Iraq account for 46.6 per cent of the proved world reserves, in contrast to 29 per cent for the United States.

Transportation inadequacies may be the source of serious petroleum shortages in some areas. If there is any serious interruption of world trade routes, tanker diversion would likely cause a shortage in northeastern United States. Should some of the more important pipe lines be converted from natural gas back to petroleum, the combined effect would be to force many users of heating oils and natural gas in the Northeast to use coal. However, with the completion of about two dozen major oil and natural gas pipe lines now being constructed, this country should be in a much better position to meet mobilization demands.

Daily operating capacity of U. S. refineries has reached 6.8 million barrels this year, and 95 per cent of the effective capacity is being fully operated. Since 1932 not only has the yield of gasoline been greatly increased, but the production of distillate fuel oils has grown at the expense of residual fuel oils.

In an all-out war the Navy will require large amounts of residual oil. Although this grade accounts for approximately 20 per cent of each barrel of crude, residual oils have been imported during much of the post-war period. In 1939 the percentage of total domestic residual sales to industry and the utilities was 27.88 as contrasted to 43.52 per cent or 241,173,000 barrels in 1950.

Discussing "Oil—Our Productivity and Resources," **C. E. Davis,** director of the Refining Division of the Petroleum Administration for Defense, addressed the Fuels Luncheon with particular reference to the impact of the war emergency on the availability of petroleum products. It is possible to deal adequately with trends in oil production and refining and their relation to crude oil resources only on a world-wide basis. There are great prospects for future discoveries of crude oil in countries outside U. S., where there are, exclusive of Russian-dominated territory, approximately 9 million square miles of sedimentary areas which have not been nearly as intensively explored as those in the U. S.

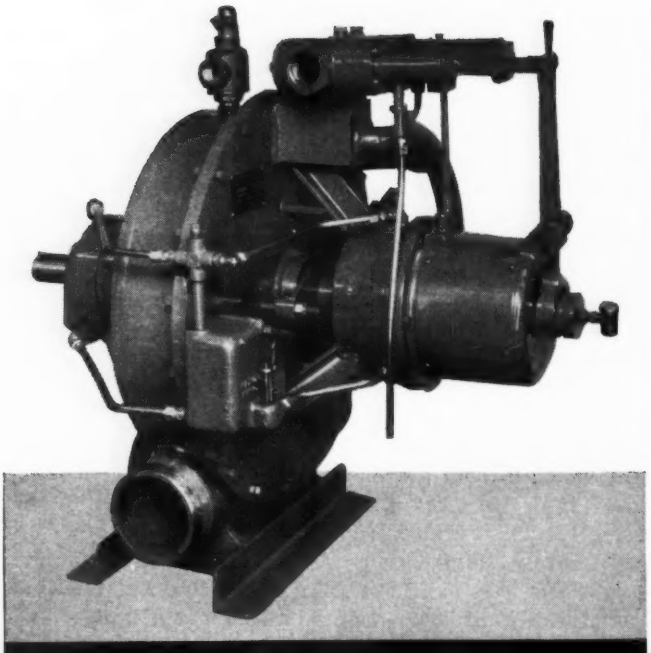
Residual oil production has not kept pace with the production of other grades because the former is ordinarily worth substantially less than the crude oil from which it is manufactured. Therefore, there is a strong economic incentive to recover a higher ratio of the more valuable products from residual fuel oil. During World War II military requirements demanded that approximately 25 per cent of the crude oil be processed to yield residual fuel, but since that time there has been a continuous downward trend. Although crude-oil runs from 1945 to 1950 increased slightly over 1,000,000 barrels a day, residual fuel oil production declined approximately 100,000 barrels a day. Under peacetime conditions it is expected that this trend will continue and that the price of residual fuel oil will find a level generally equivalent to alternative sources of competitive fuels.

Training of Engineering Graduates

A symposium on the training of young engineering graduates in industry was participated in by representatives of small, medium-sized and large companies.

G. W. Baughman outlined the engineering training program of the Union Switch and Signal Division of the Westinghouse Air Brake Co. The training period is considered as 18 months and is arranged on an individual basis so that the trainee spends a few months in each of the sections of the engineering division. Besides becoming familiar with the current problems of the engineer to whom he is assigned, the trainee is required to solve a list of prepared problems in each section. Stress is placed upon acquiring the ability to handle matters with honesty and diplomacy, and experience is provided in report writing and handling of correspondence. As the training period draws to a close, conferences are held with the section heads under whom the trainee has studied. The work assignment is then made taking into account the opinions and grading of the section heads, the desire of the new engineer, and the current needs of the organization.

How engineers can help build up a company was discussed by **E. G. Bailey,** who traced the development of engineering graduate training at the Bailey Meter Company over the past 32 years. It is interesting to note that the principle of the training course has not been changed, although it has been kept up to date in details of new equipment development. An important part of the work consists of factory assembly and calibration of meters and control units. Because many new types of equipment have been added through the years, the work in the shop has been greatly diversified, giving even broader experience to the trainee. Assignments such as taking inventory, routing parts through the shop, preparing shipping orders, and inspecting parts are made to round out the shop tour. The training period ranges between 40 and 55 weeks and includes a series of more



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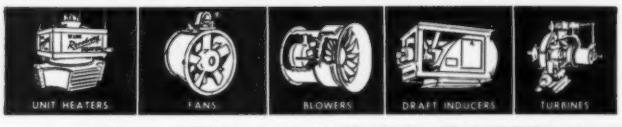

than 100 lectures covering such subjects as theory of flow measurement, principles of combustion, major steam generating equipment, electronic control, automatic control systems, process measurements, practical application problems, engineering and sales department procedures, and major company policies.

The Loop Course of Bethlehem Steel Company was described by **H. C. Houghton** who stated that the question of whether the course is to be conducted in any given year is a matter of executive determination. The program is divided into three phases: basic training for five weeks at the Bethlehem, Pa., plant; specialized training in the particular field in which the trainee is to remain engaged; and actual training "on the job." In the basic phase the trainee learns in a general way the different activities of the organization and how control of the activities is designed to create a working team. He also is given a thorough grounding in the operations of a steel plant and related manufacturing activities. Next the trainee is assigned to the activity for which he was originally hired and receives more specialized instruction. The third phase of the course is regular work on the job at which time responsibilities are assumed and the young engineer is able to demonstrate his ability to deal with actual business situations.

Industry and the Atom

The general philosophy of cooperation between the Atomic Energy Commission and industry was defined in a talk by **T. K. Glennan,** a member of the U. S. Atomic Energy Commission. In this he explained that the basic operating principle of the AEC is to accomplish its production goals through the genius of American industry, in which at present some 500 major contractors are concerned. This means that more than 100,000 persons are directly engaged in the program although only 6000 are on the AEC pay roll.

By law, nuclear energy development is a government-owned monopoly with the normal profit incentive lacking. Inventions or discoveries made in connection with the work become restricted data if dealing with fissionable material or weapons, or public property available for non-exclusive licensing if of non-military nature and declassified. Furthermore, there is the handicap of security regulations and government red tape, as well as the physical problem of health and safety.

Despite these limitations, industrial managements have readily accepted such government service. The reasons, aside from patriotism, as suggested by Mr. Glennan, are (1) that corporations are convinced that if we lose a war they will cease to exist; (2) that it is prudent to become conversant with this new technology which may widely influence the future course of engineering and production; and (3) the idea of "getting in on the ground floor" appeals to many industrial leaders.

While the fees allowed under AEC contracts are not commensurate with industrial profits that might be expected for the expenditure of a like amount of talent and effort, it is to be remembered that industry is required to make little or no financial investment in the atom business and takes no great risk of financial loss.

After-Deposits Resulting From Chemical Cleaning Process*

By P. H. CARDWELL

Dowell Incorporated, Tulsa, Okla.

THE amount and type of after-deposits which remain following the chemical cleaning of boilers are determined by the procedure used in the acid treatment and the materials employed in the conditioning or after-boil. While the formation of a protective film on the internal metal surface of the boiler is essential following the chemical cleaning, it is important that control of the amount of these after-deposits be maintained, but at the same time maximum protection of the boiler metal be obtained with a minimum amount of corrosion. Following the customary procedure for the chemical cleaning of boilers in which inhibited hydrochloric acid is employed as the cleaning medium and soda ash as the conditioning boil, there usually is an after-deposit of iron oxide present as a film upon the metal surface.

Investigation of the cause for the formation of this iron oxide film indicates that it is formed during the draining of the acid and the water rinses, following the cleaning process. While the boiler is being drained of these materials, the metal surfaces become exposed to acid and water vapors, in the presence of air. This apparently brings about the initial rusting of the metal surfaces.

The composition of this rust film, prior to the introduction of the conditioning boil material, varies according to the prevailing temperature during the drainings. In general, it may be stated that the higher the temperature at which the drainings are carried out, within the range of 130 to 160 degrees F, the greater will be the amount of magnetic black iron oxide present in relation to the other oxides of the film, such as hematite and the hydrated iron oxides. Thus, the higher the temperature of the drainings, the blacker the iron oxide film will be. Likewise, the lower the temperature at which the drainings are accomplished within the range of 100 to 130 degrees F, the smaller will be the amount of magnetic iron oxide and the greater will be the amount of gamma iron oxide present in the film. This means that, the lower the temperature of the drainings, the more reddish the iron oxide film will be. This explains why reddish films occasionally are formed following the drainings. Sometimes mixtures of red and black films will be formed.

When an alkaline conditioning boil is employed following hydrochloric acid cleaning, the gamma and hematite iron oxides are converted, either completely or partially, to the magnetic iron oxide. The extent of this conversion depends upon the alkaline material used and the temperature and time of the conditioning boil. This means that, upon completion of the acid stage and the conditioning boil, there will be present on the metal surface an iron oxide film having a different chemical composition from the original film which was formed during the draining of the boiler.

COMPOSITION AND AMOUNT OF DEPOSIT PRESENT IN BOILER TUBES AFTER CHEMICAL CLEANING

Acid-Washing Treatment						
Concentration of Inhibited Hydrochloric Acid, %	Amount of Hydrochloric Acid Consumed, %	Acid and Water Displacement Method	Conditioning Boil Material		Average Amount After-Deposit (g/lineal tube ft.)	Composition of After-Deposit
7.5	1.3	Air pressure	0.5%	soda ash	1.7	85–100% Magnetite (Fe_3O_4), 0–15% Copper (Cu)
7.5	1.6	Nitrogen displacement	0.5%	soda ash	0.2	60% Magnetite (Fe_3O_4), 20% Hematite (Fe_2O_3), 20% Copper (Cu)
7.5	1.7	Nitrogen displacement	0.25%	sodium chromate	0.5	5% Hematite (Fe_2O_3), 25% Magnetite (Fe_3O_4), 70% Copper (Cu)
7.5	2.5	Nitrogen displacement	0.25%	trisodium phosphate	0.2	40–60% Hematite (Fe_2O_3), 15–40% Magnetite (Fe_3O_4), 0–20% Copper (Cu), 10–15% Lepidocrocite Gamma (FeOOH)
7.5	2.0	Nitrogen displacement	0.1%	phosphoric acid	2.0	35–50% Magnetite (Fe_3O_4), 25–35% Unidentified Phosphate, 5–20% Copper (Cu), 5–20% Hematite (Fe_2O_3)
5.0	1.6	Air pressure	0.05%	phosphoric acid	0.35	40% Magnetite (Fe_3O_4), 20% Copper (Cu), 10% Hematite (Fe_2O_3), 10% Cupric Oxide (CuO), 10% Strengite ($FePO_4.2H_2O$), 5% Cuprous Oxide (Cu_2O), 5% Amorphous (organic)

The first five results were obtained from 450,000-lb per hr, 1400-psig boilers; while the last result was from a 750,000-lb per hr, 1525-psig unit. The first three results are comparable in that they were obtained from three boilers within the same powerhouse. Likewise, the fourth and fifth results were taken from two boilers in a second powerhouse.

* Presented at Panel Discussion on Chemical Cleaning of Boilers, Twelfth Annual Water Conference of the Engineers' Society of Western Pennsylvania. The complete papers and discussion are published in the Proceedings of the Conference which may be secured from the Engineers' Society of Western Pennsylvania with headquarters at the Hotel William Penn, Pittsburgh, Pa.

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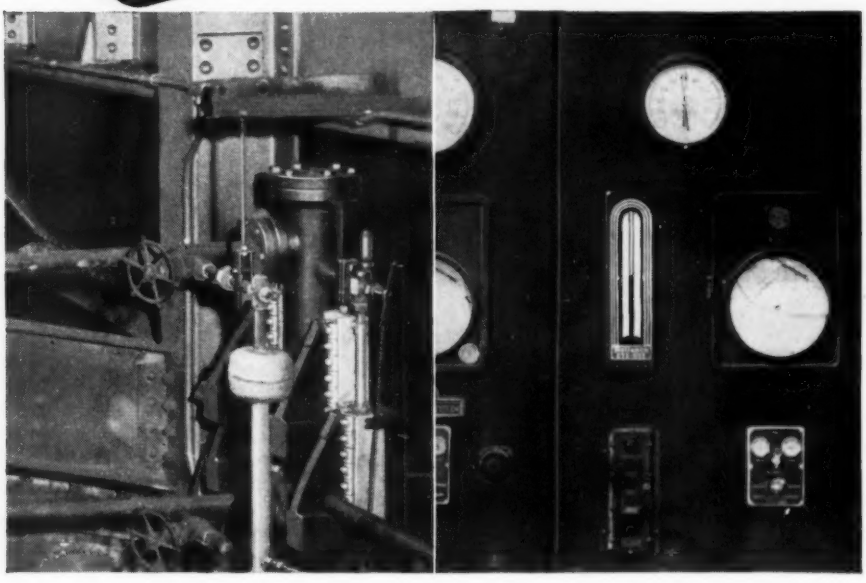

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The alkaline materials which may be used for the conditioning boil are trisodium phosphate, sodium tripolyphosphate, soda ash, sodium chromate, sodium dichromate and caustic soda. Some of these materials, such as trisodium phosphate and soda ash, act as detergents and remove a portion of the original film formed during the drainings. Therefore, when these materials are employed, there will be less film present on the metal surfaces following the conditioning boil than there was originally, after the draining of the acid and water rinses.

Acidic materials also can be used for the conditioning boil. The most common acidic material employed for this purpose is phosphoric acid. This material will form a gray iron phosphate coating upon the metal surface, providing that the acid dissolves the iron oxide which was formed in the draining procedure and that the acid is at least 50% spent (preferably 75%) before its removal from the boiler. If the phosphoric acid is drained before it has an opportunity to dissolve the iron oxide, or before it becomes 50% spent, the after-deposit which is formed will be reddish brown in color, usually composed of magnetic iron oxide, alpha iron oxide and iron phosphate. Other acidic conditioning boil substances, such as sodium acid sulfate and sulfuric acid, will leave reddish brown deposits consisting of magnetic and alpha iron oxides.

The amount of after-deposit left upon the metal depends primarily upon the amount of rusting which occurs during the draining of the acid and the water rinses. The amount of rusting at this point in turn depends upon a number of factors, the most important of which is the length of time the wet metal surface is in contact with air. The longer this period of time, the greater will be the amount of rusting.

The temperature at which the boiler is drained also is a factor in the amount of the film which is formed. The composition of the acid being drained will have some effect in that the greater the concentration of the unspent hydrochloric acid and the higher the iron content of the drained solution, the greater will be the amount of rusting. The composition of the metal of the boiler also has some influence.

The amount of rusting which takes place during the drainings of the hydrochloric acid solution and water rinses can be controlled by preventing air from contacting the metal surfaces. This usually is accomplished through the use of water or nitrogen to displace the acid from the boiler.

The preceding table gives the amount and chemical composition of after-deposits resulting from the chemical cleaning of boilers with inhibited hydrochloric acid solutions, using various means of displacing the acid and employing different materials for the conditioning boil. These results were obtained from turbine brushings of the tubes following the chemical cleaning of boilers. The information given in the table was obtained in cooperation with the various plants operated by the American Gas and Electric Service Corporation.

Symposium Discusses Boiler Instrumentation

Keen interest in problems of controls for steam power plants was evidenced by the 350 engineers who attended the Symposium on Boiler Instrumentation sponsored by the Philadelphia Section of the Instrument Society of America at the Bellevue-Stratford Hotel on November 15. Five papers were presented pertaining to control rooms, boiler instrumentation, steam temperature control, feedwater regulators, and future prospects for power plant instrumentation.

Some of the more recent problems resulting from current central station design trends were put forth by **T. Y. Mullen** of Gilbert Associates, Inc., in a paper entitled "Design of Instrumentation and Control in the Modern Power Plant." In order to have an accurate performance record of reheat units, it is desirable to meter both the primary and reheat steam flows. Since inlet and outlet pressures of the reheater vary with turbine load, pressure compensation of the flowmeter is mandatory, and the effect of fluid compressibility also becomes important because of the low-pressure ranges involved.

Initial cost of piping for a recent central station reheat installation was $161 per foot for 12-in. main steam leads and $265 per foot for the 18-in. reheated steam leads. In order to effect savings, pipe diameters are kept to a minimum with the result that higher steam velocities are encountered than in earlier installations. If orifice ratios for flowmeters are to be kept below recommended maximums, then unusually high meter differentials are found, and fluid compressibility must be taken into consideration because of the change in density of the vapor between the inlet and outlet sides. Also, greater ranges of feedwater temperatures are encountered, a typical example being a variation from 325 F at quarter turbine load to 450 F at maximum rating. Temperature compensation on the feedwater meter is mandatory if appreciable errors in this respect are to be avoided.

In the central control room the assembly of large quantities of related data at a common point poses the problem of receiving and tabulating the data and at the same time retaining a reasonable area of operation for the attendants. There have been several approaches to this problem, one of which has resulted in the development of multi-record instruments to conserve both panel board and control room space. A second approach has been the use of miniature instruments and the substitution of small diameter indicators in place of recorders. A third step has been to go back to the concept of a semi-central control room, with sub-panels at various locations in the plant. To carry this out the operator is provided primarily with indicating instruments, while recording instruments are located on the rear of the panel or at remote locations. Hydrogen control, ash and soot blowing and pump room panels are placed outside the control room, thus reducing duties of the central operators. Mr. Mullen commented on the advantages and disadvantages of each of these approaches and observed that the instrumentation for any installation is seldom duplicated even though the basic design for the plants considered may have been developed by the same engineer. Lessons learned after a period of operation serve as the basis for improvement of future projects.

Discussion

With regard to compensation of instruments it was pointed out if errors can be eliminated at time of manufacture it is not necessary to compensate for them. Even though the error is within guaranteed limits, it will always be a part of the instrument. A representative of an instrument manufacturer recommended that consulting engineering firms should have instrument specialists on their staffs so that there will be no prejudices as to what goes on panel boards. Everything else considered, graphic panels were thought to be a means of speeding up human response in times of emergency. There was some disagreement as to the effect of miniature instruments upon readability. One utility engineer spoke out against putting instruments and recorders behind a panel where they cannot be seen and wondered why there is such a demand to condense the size of the control room which is the heart of the central station.

H. C. Mittendorf of Combustion Engineering-Superheater, Inc., presented a paper entitled "What the Power Industry Designers Expect of Boiler Instrumentation," a digest of which appears elsewhere in this issue. In the discussion of this paper the need for a practical flame detector to actuate interlocks was mentioned. It was urged that boiler and instrument manufacturers set up standards for airflow measurement that are applicable to so-called standard designs and to the larger tailor-made units. One engineer asked whether a boiler efficiency meter could be designed so that it would be possible to compare the operating effectiveness of men on different shifts.

The third paper was by **E. D. Scutt** of Leeds & Northrup Company whose subject was "Modern Trends in Automatic Combustion Controls and Superheat and Reheat Controls." Today it is not uncommon to have a single combustion controller supervising the regulation of ten or more actuators, all of which must be properly proportioned to maintain uniform combustion conditions within the furnace and to follow generator load demand. For superheat and reheat controls it is also common to have a single controller regulating dampers or tilting burners during a portion of the load range and valves for the remainder of the range.

With increasing size of boiler units, combustion, superheat and reheat controls have become more complex. The only recent change in boiler design that has simplified combustion control is the pressurized furnace which eliminates the furnace pressure controller as long as induced draft fans are omitted.

With the single boiler-turbine-generator unit and centralized control room there are longer runs between the actuators and the panel, and combustion controls must be designed to accommodate these increased distances without introducing lag in the control response. To meet this requirement one control manufacturer has replaced a system of pneumatic loading pressure with an electronically balanced Wheatstone bridge circuit which is independent of distance.

Because of decreased heat storage of modern boilers, control problems are made more difficult during load changes. At the same time any response lag becomes more critical. Unless the combustion rate within the boiler can be proportioned to load changes as quickly as possible, extreme changes in steam pressure may be experienced along with other upsets in operation.

While combustion control equipment in the past has been considered the major automatic boiler control, today the combination of superheat and reheat temperature controls may require an investment practically equal to that for combustion control. In many instances control problems encountered in regulating steam temperatures are more difficult than those handled by combustion control. This is primarily due to the time lags involved and that fact that there are so many variables within the boiler which can change steam temperatures.

On reheat installations it is common practice to have at least two means of controlling one of the temperatures. This requires sequence control of two actuators from one controller. Interaction between the superheat and reheat controls must also be given consideration. On some installations it is possible to control at least one of these temperatures without affecting the other. However, where there is recirculation of hot gas both superheat and reheat temperatures are affected. In this case it is desirable to control recirculation from the equivalent of the total of the superheat and reheat temperatures and the proportioning dampers from the equivalent of temperature difference.

Discussion

It was pointed out that superheater and reheater surface are proportioned so that a minimum of desuperheating water will be necessary at some load, usually the maximum continuous rating. Although it is desirable to simplify steam temperature control, the present trend is not in that direction. Nevertheless, there are many reheat installations in operation in which the two steam temperatures are regulated within a few degrees of the control point day after day.

C. H. Barnard of Bailey Meter Company presented the fourth paper which was entitled "Modern Feedwater Control."

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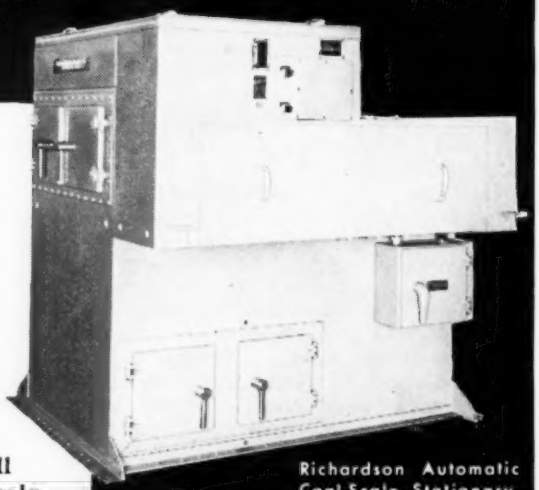

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There are four types of feedwater regulators in common use today: single-element, self-operated; single-element, pilot operated; two-element, pilot operated; and three-element, pilot operated.

The single-element, self-operated regulator is still in use to some extent on small low-pressure boilers having relatively stable flow characteristics. It has the advantages of being extremely simple in operation, low in cost, and requiring no external source of power. It has the disadvantage of undergoing severe cycling action with change in steaming rate. The addition of a pilot valve using compressed air as a control medium provides a much wider proportional band, minimizes the cycling which takes place on a change in load, and permits installation of a selector valve at the control center so that remote manual control is provided during start-up and abnormal operating conditions.

The two-element, pilot-operated system incorporates a stabilizer in the form of a steam flow controller. Steam flow acts as the primary influence to open and close the feedwater regulating valve, while the level controller is used as a readjusting control to maintain the water level in the boiler drum.

The three-element, pilot-operated regulator differs from the two-element system by the addition of a water flow meter. Primary control is from the ratio of steam flow and water flow, acting through a pilot valve having proportional band adjustment and through the Standatrol and selector valve to the feedwater regulating valve. This is basically the type of control used on large boilers for central stations and industrial plants.

The use of the reheat cycle has not added appreciable complications to feedwater control problems.

Discussion

A strong defense was made for the single-element, self- or pilot-operated feedwater regulator, and its use was advocated for industrial and utility boilers having moderate steam temperatures and capacities as high as 350,000 lb of steam per hour. It was pointed out by a service engineer that feedwater regulators are often trying to do a job for which they were not originally designed. The matter of cost of feedwater control equipment was brought up and it was stated that the control valve is the largest single item regardless of the number of control elements.

"Instrumentation for the Power Plant of the Future" was the subject of a talk by **M. J. Boho** of the Hagan Corporation. The fact that more than 60 per cent of the energy in the fuel fired is lost in the condensers will always be a challenge to steam plant designers. In a very real sense the future is unpredictable, as engineering developments take place in a zigzag and unplanned direction. Triggering impulses for new developments may be present a long time before they are set off through economic justification.

Over the next ten to twenty years the straight-through and reheat cycles will continue to be important in central stations. Combustion control systems will have to be designed with greater ranges, greater sensi-

tivity and adaptability to a combination of three or more fuels. There is considerable economic justification to develop apparatus to cut down flue gas exit temperatures, and it is quite possible that a reduction of 40 deg F below present minimum temperatures will soon be feasible. Two areas in which there is opportunity for much improvement include air-flow control and measurement and automatic control of blowdown.

It appears almost certain that instruments of the transmitting type will be used more generally, permitting several variables to be combined in one measurement.

The present tendency to duplicate automatic controls with manual controls will disappear as more confidence is gained in completely automatic systems. Further application is to be expected of automatic computing devices to give instantaneous calculations of plant efficiencies.

Second Gorgas Gasification Project Completed

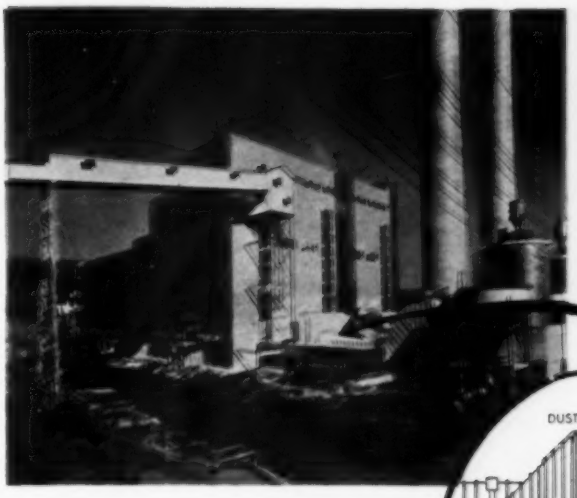
Four years of extensive experiments conducted by the Bureau of Mines at Gorgas, Ala., have shown that gases suitable for driving gas turbines and generating steam can be obtained through underground gasification, according to the second detailed report on the project just issued. Further tests may prove the feasibility of producing gases for conversion to synthetic liquid fuels or chemicals.

During one phase of the operation, two gas turbines were installed and operated for approximately 100 hr on combustible gases captured from the burning coal. The energy produced by the turbines, in turn, was used to compress air and send it back underground to the gasification system. In commercial operations, excess energy would be produced for useful purposes.

These experiments, staged jointly by the Bureau of Mines and the Alabama Power Company, were pioneered between January 21 and March 12, 1947, during which time the initial goals in gasification were attained; namely, that coal in place could be burned, and that the burning could be controlled. The early tests, however, were largely exploratory and much smaller in scope than the subsequent undertaking which commenced its first operating period on March 18, 1949, and finished its final operating period on February 7, 1951. Thereafter water flooding was started to extinguish the fire.

During the 22 months' continuous operation of the second project a total of 10,485 tons of coal, underlying almost two acres, was gasified. There appeared to be no limit to coal areas from which combustible gases could be obtained.

Over a four-month period when 65 per cent of the heating value of the coal consumed in one area was realized, the energy yield was greater than could have been obtained from the coal mineable from the same area under existing mining methods. During one 8-hr period, combustible gas with a heating value of 90 Btu per cu ft was produced at a rate of 9.4 million cubic feet a day. In other phases



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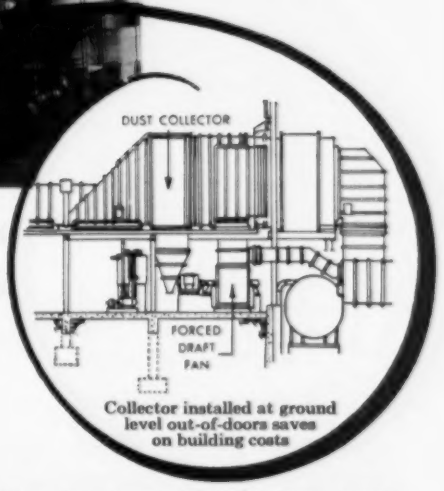

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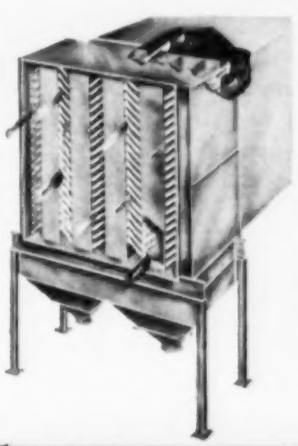

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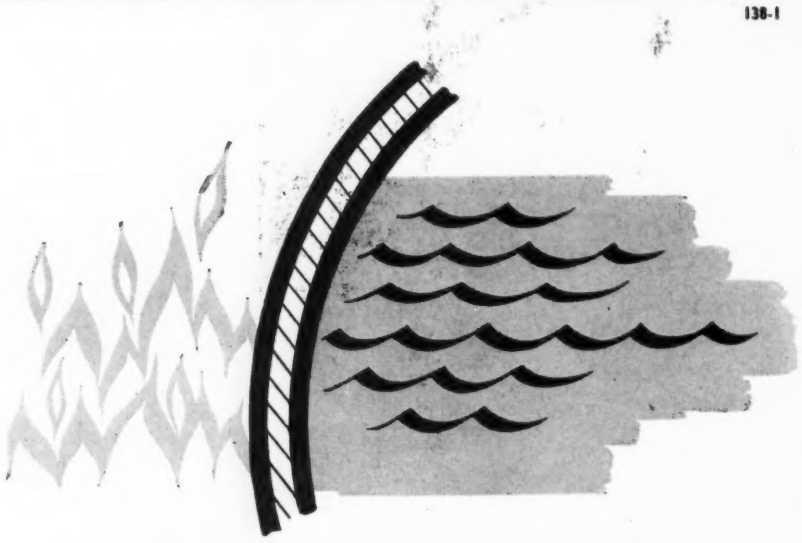

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of the experiment when new inlets and outlets were driven near the burning coal faces, gases having calorific values of 75 to 150 Btu per cu ft were produced.

Inasmuch as methods of and processes for underground gasification still are in the experimental stage, no definite economic evaluation is offered in the report. It adds, however, that provided technical aspects are feasible, underground gasification of coal can be applied for the following uses:

1. Complete combustion of coal underground with air and utilization of the heat energy in gas turbines or in raising steam.
2. Production of producer gas with air, and utilization of the combustible gas in gas turbines.
3. Production of synthesis gas by gasification of coal with oxygen and steam, and using the gas in the manufacture of synthetic liquid fuels or organic chemicals.

Further experiments are to be staged at Gorgas, involving a new installation which will feature a series of electrodes for passing electrical current through the coal seam to open passages for air and gases.

Hoover Opens Drive for Columbia Engineering Center

Former President Herbert Hoover was the keynote speaker on November 7 at a banquet at the Waldorf-Astoria Hotel, New York City, marking the opening of a campaign for $22,150,000 for a proposed new engineering center for Columbia University. The center will combine teaching, research and practice in engineering with the fundamentals of science on both the graduate and undergraduate levels. Comparable in concept to the medical center which was pioneered by Columbia nearly a quarter century ago, the new project will also encompass an Institute of Advanced Engineering Science and a Division of Cooperative Research in Engineering.

Mr. Hoover noted that engineering training had risen from the state of apprenticeship in trade or technical schools to the dignity of a university-trained profession during the course of his own life. Commenting upon the role of the engineer in overcoming evils, he stated:

"The engineer upset the canal monopoly with the railways; he upset the railways with the automobile, the airplane and the pipeline. He upset the anthracite monopoly with coke; he upset kerosene oil with the electric light, and he assured that most of the streams would remain as scenery by making cheaper power with steam."

With regard to the current shortage of engineers, Mr. Hoover observed that a young mechanic with three years of training can earn more take-home pay than a young engineer with six years of training and three years more of experience. But more engineers are needed to carry on the nation's work. One way to overcome the present rearmament burden is by the increase of productive power through new engineering developments.

REVIEW OF NEW BOOKS

Any of the books here reviewed may be secured through Combustion Publishing Company, Inc., 200 Madison Ave., N. Y.

Water Treatment

By Eskel Nordell

For one desiring a comprehensive survey of the problems and processes of water treatment for industrial and domestic uses, this book should prove most valuable. Composed of 18 chapters and more than 500 pages, it provides an exhaustive, practical and up-to-date reference work for executives, consultants, engineers, chemists, operating personnel and students interested in water treatment.

The opening chapter on industrial water supplies has a clear explanation of methods of expressing water-analysis units. Included also are tables of average analyses of 98 river waters, 50 lake waters and 50 spring waters.

Succeeding chapters discuss dissolved mineral matter, dissolved gases, turbidity, sediment, color, organic matter, tastes, odors and microorganisms. A chapter on water requirements lists more than 300 users of water-treating equipment and the types of equipment required by various industries.

The different processes and equipment used in treating water are covered in detail in eleven chapters. The appendix section contains forty-three tables of conversion factors and equivalents and three curves which will be found useful in making calculations.

Many of the chapters have separate bibliographies of appropriate publications and articles on water-treating materials, processes and equipment. An interesting feature of the book is the use of informal notes at various points in the text, where the author relates incidents based on his personal experience and introduces other information indirectly related to the water-treatment field. In the chapter on boiler feed waters there is a particularly pertinent note on what are termed "illegitimate boiler compounds."

The author has spent more than 30 years in the water-treatment field, and has written a work that gives every indication of intensive knowledge of his specialty. The book contains 526 pages and sells for $10.

Fuel Oil Manual

By Paul F. Schmidt

For the oil user who wants to know about fuel oil selection, properties, handling, burning, troubles and remedies this manual should prove quite useful. Written by a chemist with many years' experience in the oil industry, it provides a working knowledge of fuel oil and how to use it. Since the manual is directed primarily to the purchaser of fuel oil and salesmen for that product, emphasis is largely on the practical side of refining, blending and combustion. No attempt is made to go into the specialized details of the refining process or to delineate in more than a general way tests that may be performed to determine fuel oil properties.

Introductory chapters discuss briefly the chemistry of petroleum and petroleum refining processes. After an explanation of the grades and types of fuel oils, such properties or components as gravity, heat of combustion, viscosity, water and sediment, carbon, ash and salt residue, flash and pour points, sulfur, and color are examined. In many cases typical commercial tests are mentioned, and typical ranges of the extent to which the various components are found in fuel oil are listed.

Concluding chapters take up the preheating of fuel oils, stability, treatment by means of additives, reclaimed fuel oils, blending, transportation and storage, typical specifications, and a list of 22 common fuel oil troubles.

This 160-page manual fills the need for a practical book on the principal properties of fuel oil that are of concern to the purchaser of fuel oil and the operator of equipment burning fuel oil. It is not an exhaustive scientific treatise, but it is a simply written, concise manual which answers many practical problems. The manual sells for $3.50.

Industrial Piping

By Charles T. Littleton

In the chemical and process industries of today literally scores of different materials and types of piping are required. To use these it is necessary to have a collection of catalogs, often as many as a hundred or more, for reference purposes. This book is a compilation of data contained in such catalogs so that a rational choice may be made between one material and another.

The intention of the author is to incorporate only that information which will be required by practicing piping designers, erection foremen and engineering students. For that reason no attempt has been made to cover a theoretical presentation of hydraulics, thermal expansion and radiation. On the other hand, practical methods commonly in use for solving such problems as the sizing of pipes and determining pressure drop are presented.

The book explains the procedure for designing and erecting the piping in a modern industrial plant from the preliminary flow sheet and plot plan to the final erection drawings. Individual chapters are devoted to special problems encountered in these services: steam, water, oil, and gas, air and instrument piping. The introductory chapter contains a table of valve equivalents for valves in the pressure classes up to 600 lb. These equivalents give the catalog numbers of about a dozen leading valve manufacturers.

There is a chapter on the estimation of piping costs prepared by R. A. Dickson. Sixty-five examples, with accompanying charts and tables, are worked out, based on N factors. This N system has been used by Mr. Dickson for a number of years and has appeared in part as articles in issues of *Chemical Engineering*.

The author of this useful work on industrial piping was formerly in charge of piping for the American Cyanamid Co. Containing 394 pages, the book sells for $8.

Thermodynamics Fourth Edition

By A. W. Porter

Many books under this title are more or less voluminous and quite technical. By contrast, this little pocket-size book, $4^1/_2 \times 6^1/_2$ in. and containing only 124 pages, serves as an introduction to the subject by covering the fundamentals in a readily understandable manner. Among the items explained are the beginning of mechanics, measurements of heat and work, the laws of thermodynamics, reversible and irreversible operations, entropy, equilibrium, etc.; and of special interest are comments on the work of Carnot, Lord Kelvin, Maxwell, Joule, Regnault, Clausius and others whose thinking and experiments formed the basis of our present knowledge of the subject.

Perusal of the booklet is recommended especially for those who desire to brush up on their knowledge of thermodynamics or those who may desire to acquire a general knowledge of the subject. It is not claimed to be a substitute for a more complete treatise, nor a textbook.

The price is $1.50.

Industrial Furnaces, Vol. 1 Fourth Edition

By W. Trinks

The author, who is now professor emeritus of mechanical engineering at Carnegie Institute of Technology and widely known as a consultant and authority on industrial furnaces, brought out the first edition in 1923, with subsequent editions appearing in 1926, 1934 and 1951. These have long been accepted as standard for design.

The present edition amplifies the basic principles underlying industrial furnace design and operation by incorporating the latest facts and data on heat transfer as brought out by research and present-day operating technique. About 40 per cent of the text has been rewritten and a large number of new illustrations added.

Following the introductory chapter which covers the use of furnaces, subsequent chapters deal with the heating capacity of furnaces, their fuel economy, heat-saving appliances, the strength and durability of furnaces, and movement of gases within them.

There are 526 pages and the price is $10.

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New Catalogs and Bulletins

Any of these may be secured by writing Combustion Publishing Company, 200 Madison Avenue, New York, 16, N. Y.

Boiler Service Valves

A 22-page bulletin, E125, on valves for boiler services, such as surface blow, bottom blow-off and water-column blow-off connections, has been published by the Everlasting Valve Co. It describes quick-opening and slow-opening straightway valves, angle valves, "Y" valves, and duplex blow-off units, with specifications, materials of construction and dimensions of each type. Illustrations include details of design, sectional and exploded views, and explanations of valve operation.

Combustible Gas Analyzer

Instrumentation Data Sheet No. 10.15-4a prepared by the Brown Instruments Div. of Minneapolis-Honeywell Regulator Co. describes a new thermocouple-type analyzer which uses Brown ElectroniK potentiometers for combustible gas analysis. Built by Davis Emergency Equipment Co., the instruments may be used in connection with hydrogen cooling systems for generators. The four-page data sheet explains the operation, application and design features of the new analyzer.

Liquid-Level Gages

Bulletin No. 176 prepared by Jerguson Gage and Valve Co. describes and illustrates Truscale remote-reading liquid-level gages. A schematic diagram explains how the gages are connected for measuring water levels in boiler drums. Typical specifications are also included in the eight-page bulletin.

Flow Signal Transmitter

The Hagan flow signal transmitter is described in an eight-page bulletin, No. 2551, issued by the Hagan Corporation. Diagrams in the bulletin illustrate seven suggested uses of the transmitter in measurement of flow, liquid fuel, liquid level and absolute pressure. A cutaway drawing illustrates the principles of operation.

Power Plant Equipment

A 12-page bulletin describing in condensed form the complete line of Swartwout power plant equipment has been made available by The Swartwout Co. Printed in two colors, the catalog is illustrated with photographs, flow diagrams, dimensional drawings, performance charts and specification tables. It describes control valves, pressure and temperature master controls, water regulating valves, liquid level and drainage controls, differential pressure controls, feedwater heaters and regulators, air separators, exhaust heads, and other accessories.

Oxygen and Hydrogen Analyzers

A four-page bulletin, No. 148 BP, describing and illustrating dissolved oxygen and dissolved hydrogen analyzers for steam generating plants has been prepared by the Cambridge Instrument Co., Inc. The bulletin includes a schematic diagram of the oxygen analyzer, a typical record of dissolved oxygen content in feedwater, and statements concerning the ranges and accuracy of the instruments.

Turbine-Generators

Design and construction features of Allis-Chalmers WA-series steam turbine-generator units manufactured in NEMA ratings of 2000 to 7500 kw are described in Bulletin 03B7654 released by that company. The turbines are built in the condensing, non-condensing, and automatic-extraction types. Cross sections of each type, the complete unit, and governing valves and operating mechanism are shown in the eight-page bulletin.

Water Columns and Gages

A 20-page bulletin on water columns and gages, WG-1811 (1951), has been issued by the Yarnall-Waring Co. In this revised edition improvements in flat-glass inserts used in Yarway high-pressure gages are described. Construction details of water column gages, typical arrangements, a spare parts list, dimensions and prices, and photos of representative installations are included.

Ash Handling

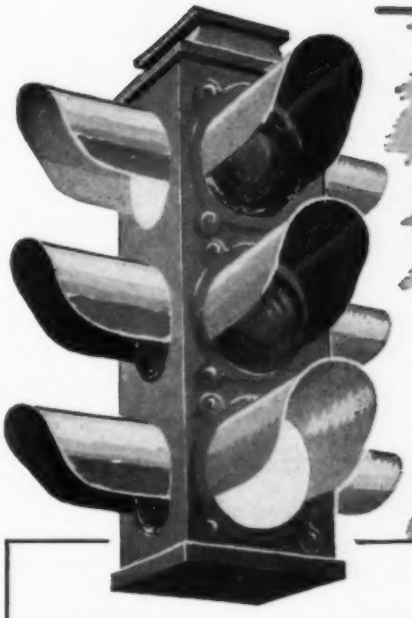
The Allen-Sherman-Hoff Co. has prepared a four-page bulletin entitled "Modern Ash Handling for Modern Power Plants." Four recently completed power stations which have hydraulic and pneumatic ash-removal systems are illustrated.

Coal Handling

A 44-page booklet, No. 2410, on coal and ash handling equipment for power plants has been published by Link-Belt Co. Subjects taken up include coal unloading, auxiliary unloading, coal storage and reclaiming, crushers, transporting and distributing equipment, ash handling and water intake screens. There are several useful layout drawings and many interesting installation photographs.

Dust Collection

Buell Engineering Co., Inc., has prepared an attractive 28-page bulletin entitled "The Collection and Recovery of Industrial Dusts." Following an explanation of what dust is and techniques for its analysis, the bulletin mentions factors influencing the choice of dust-collecting equipment. A number of types of mechanical and electrical collectors, together with their principles of operation, are described and illustrated. Typical applications are shown for such industries as cement and rock products, petroleum, pulp and paper, chemical, and food as well as for fly ash collection.



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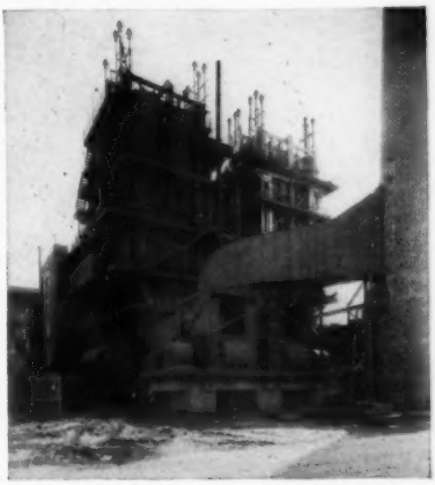

The Line of Green Induced and Forced Draft Fans includes many types in many sizes which are described in Bulletin No. 168. We shall be glad to discuss your fan requirements with your own engineers, with your consulting engineers and with the boiler manufacturer of your choice.

THE GREEN *Fuel Economizer* COMPANY INC.

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Business Notes

Dowell, Inc., Tulsa, Okla., has promoted A. C. Polk, Jr., from assistant manager to general manager of the Company.

Canadian Johns-Manville Co. Ltd. has named Karl V. Lindell vice president and general manager of the Company's Asbestos Fibre Division.

R. E. Moyer, Inc., is the name of a new firm in Catasauqua, Pa., organized by R. E. Moyer and L. W. Master to manufacture and fabricate power plant and process equipment of alloy and steel plate.

The Economy Pump Division of the Hamilton-Thomas Corp., Hamilton, Ohio, is being consolidated with the **C. H. Wheeler Mfg. Co.** of Philadelphia, another division of the Corporation. Production, engineering and pump sales are being moved to Philadelphia, and Economy pumps will henceforth be known as Wheeler-Economy pumps.

Combustion Engineering-Superheater, Inc., New York, has concluded a "License and Technical Assistance Agreement" with **Fabrica de Caldeiras A Vapor, "Cyclope,"** S.A., of Sao Paulo, Brazil. This is the fifth such agreement to be concluded by the Company in the past two years; the others relate to France, Italy, Argentina and Japan. Under the agreement Cyclope will be licensed to manufacture Combustion's line of standard water-tube and fire-tube boilers, as well as several types of stokers, and will receive technical assistance. The Company's subsidiary in Brazil, **Combustion Engineering Ltda.,** with offices in Rio de Janeiro and Sao Paulo, will collaborate with Cyclope and handle directly such large power station contracts as have to be engineered and partially manufactured in the U. S. Even on such projects, however, Cyclope will supply as much of the material and labor as possible.

Baldwin-Hill Co., Trenton, N. J., has acquired the Rockdale Insulation plant, near Housatonic, Mass., from the Eagle-Picher Co. This plant will be operated to develop new products as well as to manufacture and distribute materials to the New England territory.

Western Precipitation Corp., Los Angeles, has opened a new branch office in Atlanta, Ga. This will be in charge of Kenneth H. Cree, who has had some sixteen years' experience in the recovery of dust, fly ash and kindred products by electrostatic and cyclonic methods.

Bigelow-Liptak Corp. celebrated its 25th Anniversary by moving its general offices to larger quarters at 2550 W. Grand Blvd., Detroit, Mich.

Warren Steam Pump Co., Inc., Warren, Mass., announce that on and after January 1, 1952, all business formerly transacted through the office of Parkman A. Collins Co., 75 Federal St., Boston, will be handled by Warren Steam Pump Co., at 6 Leonard St., Belmont, Mass.

Personals

John J. Forbes has been appointed director of the U. S. Bureau of Mines. He had been chief of the Bureau's Health and Safety Division since 1948.

John F. McKee was recently awarded a Certificate of Merit by the Franklin Institute, Philadelphia, for his development of the small light-weight Yarway steam trap.

George A. Fricke, for many years identified with the Griscom-Russell Co., has recently become manager of the Heat Exchanger Division of Superior Combustion Industries, Inc., New York.

Dr. A. C. Fieldner, chief fuels technologist of the Bureau of Mines, received an honorary award at a banquet on November 28 from the Washington Society of Engineers.

Walker L. Cisler, executive vice president of The Detroit Edison Co., succeeded to the presidency of that company on December first upon retirement of James W. Parker, who will continue in a consultative capacity. Before becoming associated with The Detroit Edison Co. in 1943, Mr. Cisler had been executive engineer with the Public Service Electric and Gas Co. of N. J. During and following World War II he was associated with the War Production Board, the occupational forces in Germany as a colonel on General Eisenhower's staff and later as a consultant for the ECA.

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The efficient Sauerman Power Drag Scraper stores maximum tonnage in any available space—quickly reclaims coal from any spot in the storage area—all under the "push-button" control of a single operator. Added safety is a feature too, because the scraper builds a compact pile that helps prevent spontaneous combustion. Installation cost is low. Maintenance expense averages but a fraction of a cent per ton handled. For the simplest, most profitable way to stockpile coal—insist on a Sauerman System!

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